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Storage of Very Cold Water in Stratified Cool Thermal Storage Tanks

Report No. 91-304-K

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ABSTRACT

Cool thermal storage is an effective demand management tool that can reduce the summer peak electrical load. This report summarizes the results of a laboratory study to investigate the effects of introducing water below 4°C to a cool stratified water storage tank in an effort to increase its thermal storage capacity while maintaining an acceptable stratification. Tests were conducted using two standard types of inlets, three different charging strategies, and very cold inflow temperatures. The results showed that though mixing does occur with the very cold water inlet temperatures, an increase in storage capacity (1.5% to 13.5% — depending on the strategy used) is achievable. However, due to mixing, these results fall short of initial maximum anticipated capacity increase calculated at approximately 35%. A detailed review of the experiments and the results are included in the Appendix.

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Storage of Very Cold Water in Stratified Cool Thermal Storage Tanks

INTRODUCTION

Cool thermal storage is a very effective demand management tool that can reduce the summer peak electrical load. Although Ontario Hydro still experiences a system winter peak load, slowly changing global weather patterns and the increasing market commonplace of residential and commercial air conditioning systems may soon shift the peak to the summer.

Stratified water cool thermal storage systems operate by chilling a fixed supply of water (usually at about 15°C), during off-peak period (presently from 11:00 p.m. to 7:00 a.m. as set by Ontario Hydro), to a temperature of 4°C. This process is known as 'charging'. Due to the economics involved, only one tank is usually used to contain the water before and after chilling. It relies on the naturally stratifying effect of water to keep the two temperature regimes separated [1]. The charging occurs during the entire off-peak period until the start of the on-peak period, at which time the tank is switched to a 'discharge' mode. In this mode the chilled water is drawn from the tank, used to supplement the daily cooling requirements of the building, and is then returned to the tank at a warm (15°C) temperature.

Stratified water storage technology directly competes with two other cool storage technologies; these are the ice harvester and phase change systems. The primary advantage of these systems is their utilization of a phase change (of water or other medium) to store energy. These systems use approximately one fifth the volume required by a stratified water storage system to store the same amount of energy. However, they have a far greater capital cost, are much more complex, and are more difficult to operate.

It is known that when using chilled water storage tanks for cool thermal storage systems, a tank with temperature stratification is more efficient and has a greater capacity for load shifting than any mixed condition. Present tanks are designed for maximum stratification to provide the best capacity. This capacity is however limited by the charging inlet temperature of 4°C, which represents the maximum density point of water. Water introduced to the bottom of a tank at a temperature lower than 4°C will tend to rise and cause mixing. If mixing can be prevented, or at least minimized, then an inlet temperature reduction could increase the thermal capacity of a tank, therefore allowing more load to be shifted to off-peak hours. Consequently, a study was conducted to investigate means of storing the water as low as 0.5°C while still maintaining an acceptable stratification and proper operating conditions. If technically feasible, the benefits are direct, for a drop in inlet temperature to 0.5°C could increase the tank capacity by 35%. In consideration of the estimated 24 million litres of stratified water storage presently installed in Ontario [1], this could allow an extra 3.2 MW of shifted capacity in retrofit alone.

OVERVIEW

Experiments investigating the feasibility of low temperature stratified water storage charging were conducted on a scale model of a stratified water storage tank. The tank was constructed of glass and lined with foam insulation to minimize heat transfer through its surface. An analysis was done to ensure similarity between the model and an actual full scale tank. As a control, standard conditions were modeled and the tank was charged with water at 4°C at a predetermined flow rate for eight hours.

A series of experiments were then completed where very cold water at approximately 0.5°C was generated and introduced into the tank under a number of configurations, but at the same flow rate and for the same period of time. The results were then compared to the control experiment to determine any possible increase in the thermal storage capacity of the tank.

The experiments encompassed two different types of inlets and three different charging strategies. A standard bottom and side inlet were tested with the bottom inlet achieving a 3.5% capacity increase over the common side inlet. The bottom inlet was then used for the charging strategy tests. Charging the tank for the specified time at the predetermined flow rate with very cold water achieved an additional capacity increase of 10% (for an overall increase of 13.5%). Installation of horizontal screens increased capacity by 5%, up to a maximum of 12% with ten screens. Finally, reversing the inlet with the outlet (ie, introducing very cold water at the top of the tank) when the mean tank temperature reached 4°C gave an additional capacity increase of 1.5%.

Experiments were also done to ensure that after charging, the tank was able to provide a constant flow of 4°C water from the bottom during its discharge cycle.

CONCLUSIONS

The experiments, within the scope of this project, demonstrated that it is technically feasible to introduce water below 4°C for charging a stratified storage tank and realize some increase in its thermal storage capacity.

The resulting capacity increases achieved are smaller (~30%) than those originally anticipated at the onset of this study. Some mixing of the very cold water does occur when it is introduced into the bottom of the tank which correspondingly diminishes the capacity. Based on a projected 13.5% increase by utilizing the technology discussed in this report, the retrofit potential for the existing stratified water storage systems in Ontario is estimated at 1.4 MW. However, this technology reduces the size of the storage tank required and therefore could generate a more favourable economic scenario for new building developers who must decide for or against a cool thermal storage system.

RECOMMENDATIONS

Further research should be conducted on the various filling strategies and their effect on the stratification and storage process of the tank. Additional tests using a larger capacity tank and a more detailed method of determining the internal convection flows and temperature stratification would result in an improved picture of the mixing phenomenon and allow better conclusions to be drawn.

Since the capacity gains realised were relatively modest, work should be done on the economics of increased tank capacity versus the reduced chiller efficiency of producing water below 4°C.

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[1] Reid, Crowther & Partners, Ltd.; George Reeves Associates, Inc., *Stratified Chilled-Water Storage Design Guide*, EPRI Report No. EM-4852, Electric Power Research Institute, May 1988.

APPENDIX

STORAGE OF VERY COLD WATER IN STRATIFIED TANKS

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1. INTRODUCTION

Stratified thermal storage tanks are used in air conditioning systems to provide large volumes of cold water to absorb heat during peak demand periods. The water is cooled by chillers which either operate continuously or only during off-peak hours. The capital cost of the tank is offset partly by the savings in the capital cost of the chiller equipment. Savings are also realized by reduced charges for peak load power since the chillers operate during the night.

In most installations, cost dictates the use of a single tank to store both warm and cold water. At the start of a cycle the tank is full of warm water at ~ 15° C. Chilled water, the density of which is larger, is introduced through an inlet system at the bottom of the tank to minimize mixing with the warmer water. The design parameters are well known for water in the temperature range from 4° C to 15° C because the density of water decreases steadily with increasing temperature. Between 4° C and the freezing point at 0° C water density increases. Introducing cold water along the bottom of the tank inhibits mixing because it is less buoyant than the water above. The operating range is thus 11° C. This defines the thermal capacity of the tank which is the volume multiplied by the difference in mean temperature betweem the warm and cold conditions. The operating range cannot be extended at the upper end because of the need for a temperature difference in the cooling devices and is not extended below 4° C because of perceived problems with mixing in the tank. Chillers can operate effectively at temperatures as low as 0.5° C but this water is lighter than that at 4° C. If it were pumped into the bottom of the tank it would rise and mix with the surrounding slightly warmer water to produce 4° C water. Mixing of water in this zone of density inversion is not well understood. If it could be predicted then the thermal capacity of tanks could probably be increased. For example, if the cold water could be stored at 2.5° C this gives a range of 12.5° C which represents capacity increase of about 15%. If stored at 0.5° C the increase would be about 35%.

In practice the demand for cold water in greatest in the middle of the day but is appreciable up to late in the evening. These are the peak hours for power demand so the recharging must be accomplished during the 8 hours when off-peak power is available for the maximum of savings. It is usually found in the economic design of a system that the provision of thermal capacity for

the hotest days on record requires an excessively long pay-back period. The capacity requirement is set by a weather condition that will be occasionally exceeded and it is accepted that during these periods the chillers will operate during some of the peak hours.

2. BACKGROUND

Studies of tank operation with water above 4° C have provided a clear picture of the mixing of warm and cold water and the formation of the interface between the two. Figure 1 is a sketch of the tank showing the side inlet most commonly installed. During the filling with cool water, a jet issues from the inlet and flows across the bottom. Turbulence in the jet entrains water from above and mixes it with the cool water. The distance from the inlet to the tank wall is very large compared to the height of the inlet so the turbulence dies out and mixing ceases before the jet flow is reflected at the wall. This action defines a mixing layer which develops into the thick interface between the warm and cool water sketched on Figure 1. The useful capacity of the tank is the volume below the interface which is at the temperature of the inflow. An ideal inlet design would produce an inflow which was uniform over the cross-section and moved slowly upward with a velocity = Q/A. The interface thickness would be zero at the start but it would grow with time because of heat conduction. For the tanks used in most buildings the thickness after 8 hours growth would be less than 30 cm under this ideal inflow. This would not represent a serious loss in thermal capacity of a tank since it is about 1.5% for a 10 m deep tank. However, it is possible to have intensive mixing from the common design of inlet which can produce an interface which contains 10% of the capacity.

If the temperature of the inflow is less than 4° C, then the temperature field would develop differently from that described above. Consider a tank filled with water at 15° C but with an inflow at 0° C. Assume that there is mixing in a layer at the bottom and formation of an interface. The temperature below it would be 4° C which is the maximum density point. The 0° C water on entry would be lighter than its environment and therefor would rise. If it rose unmixed to the bottom of the interface, it would there combine with 8° C water which is of the same density. The result is 4° C water. This simple picture of the tank would lead to the conclusion that it is not possible to store water at temperatures less than 4° C in normal operations. However, there are uncertainties in this picture which may be utilized to produce a mean temperature which is indeed

less than 4° C. Furthermore, the mixing of 0° C and 8° C water results in a larger volume of water at 4° C than if the inlet flow were 4° C. There is no loss in thermal capacity but the volume increase has to be considered in the design and operation of the tank.

3. OBJECTIVES

The objectives of this experimental and theoretical study were the development of design and operating schemes to increase the thermal capacity of chilled water storage tanks by reducing the thickness of the interface and producing a mean temperature of less than 4° C below the interface. To achieve the objective the following ideas were studied:

- mixing of the cold water in the layer below the interface may not be intense and so the temperature at the bottom could be less than that at the interface. Two inlet systems were tested to evaluate the effect of the design on the intensity of mixing.

- a horizontal screen can be designed to prevent interchange of the water volumes above and below it. Lighter, colder water could be kept below the screen while the tank is being filled.

- use can be made of the outflow system at the top of the tank as an inflow for the lighter water. The scheme would allow the tank to fill with water at 4° C from an inflow at 0° C and then reverse the flow system to put the 0° C water on top of the 4° C water. This is a stable configuration.

4. EXPERIMENTAL METHODOLOGY

A glass wall tank about a half meter high was used in all tests. It was lined with foam insulation to minimize heat conduction from the warm to cold layers. A rack of thermocouples was constructed so that temperature profiles over the full depth could be determined at any instant during tank filling. The tank was then operated in a routine similar to that in a full-size tank. The water temperatures were exactly those used at full scale and the velocity from the inlet was scaled to produce the same inertia to the buoyant force on the water particles. Appendix A

contains a full discussion of the rules for similarity and how these were satisfied in the laboratory.

The tests consisted of starting the inlet flow and observing the temperature profiles as the interface developed. Details are provided in Appendix B. The resulting profiles were plotted and compared to those calculated from the theory of heat conduction. The difference between these defined the mixing in the interface and the cold region below it. Thus the physical mechanisms produced by the inflow of very cold water could be explained and used to develop ideas for the storage of water at temperatures below 4° C.

All of the test results were coordinated by theory and ideas can thereby be extended to full scale tanks. However, it is realized that there may be unexpected phenomena in these units which are 10 to 50 times the size of the laboratory tank. Measurements should therefore be made of the temperature profiles in operating tanks supplied with very cold water to verify that the theory is truly applicable.

5. RESULTS

A detailed description of the test results with the significant measured temperature profiles are contained in Appendices at the end of this report. These also contain some of the theoretical analyses used in the interpretation of the results.

The first tests were aimed at an understanding of the flow produced by the commonly used inlet system. A slot inlet was installed at the bottom corner of the tank which directed the flow horizontally along the bottom as sketched in Figure 2. The test employed salt water as the heavy fluid to simulate cold water and fresh water simulated the warm water. Figure 3 presents 4 of the photos taken of the developing pattern. The first one was taken just after the salt water flow was started. Dye was introduced into the inlet for the first few seconds of the test and it can be seen to be distributed uniformly though a mixing layer. This was a turbulent zone produced by the shear between the high velocity inflow jet and the stagnant water in the tank. The thickness of the mixing layer was approximately h. The three succeeding photos show a distinctive dye pattern. There was an intense line of color which was at the top interface. This line was the top of the mixing layer at the start. Measurements of its position revealed that there is no flow across

it because it was moving up at a velocity Q/A where Q is the discharge from the inlet, i.e. the volume divided by the filling time and A is the horizontal cross-section. Below the interface there was a clear zone with a curved edge on the inlet side. Particles could be seen to travel across the bottom of the tank, rise up the opposite side wall and then return in a curved line to be entrained by the jet. These define the streamline pattern on Fig. 4. The curved line separates a zone on the left in which the particle motions were very small. There was a slow inflow to this zone at the inlet which means that the density in this zone is less than that in the larger zone to the right. When the inlet flow was stopped the curved line disappeared as the lighter fluid flowed over the heavier and blended into the interface. Density profiles and the analysis of the interface are described in more detail in Appendix C. These demonstrated that the inlet flow matched that found in previous work on model and full size tanks.

The development tests described in Appendix D were next undertaken to determine the accuracy obtainable in the model with warm and cold water. These evaluated the measuring system and determined the thermal losses. It was found that there were small losses but results could be corrected accurately.

Tests with very cold water were conducted with both the common side inlet and with a more complex bottom inlet. These are described in Appendices E and F. The circulation pattern produced by the side inlet shown in Fig. 4 was observed to be an effective mixing mechanism so the temperature below the interface was uniform at 4° C. Consequently the interface rose faster than the particle velocity Q/A. This meant that the cold inflow mixes with the warm water at the interface giving the thicker cold zone. However, it also meant that the interface thickness was reduced by the mixing process and thus was thinner than in the case of 4° C inflow.

The bottom inlet shown on Fig. 2 was designed to produce a uniform small upward velocity. The two features were a pattern of jets directed against the bottom and a horizontal plate to smooth the high jet velocities. The results demonstrated that the recirculating pattern was not present but that there was mixing produced by the unstable buoyancy. That is, the colder but lighter water mixed continuously as it rose so there was a temperature gradient over the layer below the interface. This gradient represented a 0.36° C reduction in the mean temperature which amounts to an increase in thermal capacity of 3.6% There is an additional benefit from the bottom inlet

which also offsets the additional cost. Previous tests demonstrated that the interface is thinner with this inlet. The buoyancy driven mixing also reduced the thickness so there is an additional increase in capacity of a few percent.

The effect of a horizontal screen as a storage device for lighter but colder water was studied in a series of tests described in Appendix G. The screen separates the liquids only if the pressure loss by the upward flow is larger than the buoyant force produced by the heavier fluid. That is, there is an upward force on the water equal to the drag of the screen. This means that the separation occurs only when there is an upward velocity. When the flow stops the fluids will mix. A test demonstrated that the temperature below a screen is close to that from the inlet and thus a theory can be developed for the placement of one or many screens at any elevation. The theory shows that very cold layers can be created in the lower levels but the important parameter is the mean temperature which will result when the inflow stops. One screen at an elevation of 0.1 of the height produces a capacity increase of 6% and the increase is reduced as the height is raised. Ten screens produce an increase of 12%.

The next scheme studied was the control of the filling operation. It is assumed that there is chiller capacity to provide 0° C for the duration of the filling time which is normally 8 hours. If a side inlet is installed then the tank will be filled with 4° C water after (15-4)° Cx8 hours /(15-0)° C = 5.9 hours. Now if the 0° C water is pumped into the bottom, the tank will mix continually and the final temperature will be 3.06° C. This is a capacity increase of 10%. However, if the inflow and outflow systems are interchanged as is done at the start of the discharging cycle then lighter water will be introduced above the heavier water and the mixing should be less. A test in which this procedure was simulated is described in Appendix H. The bottom inlet design was used for the upper unit to minimize mixing and the measured increase in capacity was 11.5%. This is an improvement of 1.5% over the continuous filling from the bottom. In a full size tank there should be relatively less mixing so the increase should be larger. The maximum possible increase is 26%.

A question which has not yet been addressed is the temperature regime which will be produced by the refilling with 15° C water of a tank of 0° C water. Will the mixing produce large scale motions that will lead to masses of water at an intermediate temperature? The coldest possible

temperature is desired for the operation of cooling units. This was investigated by filling the tank with 0.9° C and recharging with warm water from the top in a test described in Appendix I. It was found that the tank temperature was uniform over the depth at all times and rapidly approached 4° C. The test was repeated with a layer of ice on the surface and again the outlet temperature was less than 4° C for a very short time. These observations have consequences in the operation of any tank. The coldest reliable temperature from a tank is 4° C regardless of the storage temperature. The outflow volume of 4° C water is consequently larger than the tank volume.

6. CONCLUSIONS

The objective of this study was the development of techniques for the storage of water at temperatures less than 4° C in a stratified tank. The approach was a laboratory scale simulation and analysis of measured temperature profiles. The effectiveness of a scheme was evaluated by the magnitude of the mean temperature of the tank after the filling time, that is, the volume divided by the discharge. Analysis of the results indicated that for an inflow at 0° C:

- The standard side inlet design produces a recirculation which generates a layer at 4° C that fills the tank after 0.6 of the filling time. Continuing the inflow at 0° C results in an increase in capacity of 10%.

- The bottom inlet design in which the flow is distributed over the cross-section and is directed at the floor of the tank produces less mixing and an additional increase in the capacity of 3.5%.

- Horizontal screens can prevent mixing and so storage of cold water underneath water at 4° C can be achieved. One screen can provide a capacity increase of 5% over the gain from a bottom inlet and 10 screens can provide an increase of 12%.

- Continuous inflow at 0° C from a bottom inlet for the duration of the filling time results in a mean temperature of 2.9° C which represents a capacity increase of 1.5% above that for a side inlet. If the inflow is switched to the top inlet when the interface reaches the top

the mean temperature is reduced a additional 0.13° C which represents a capacity increase of 1.5%.

- Addition of ice at the top at any time after the mean temperature has reached 4° C provides an additional increase in the thermal capacity of the tank. The increase is proportional to the volume of ice. The capacity increase is effective when the tank is refilled with warm water. The outflow at the bottom is steadily 4° C while the ice is melting and while the interface descends to the bottom. The interface starts from the top when the ice is completely melted.

- Refilling of a tank at 0° C with 15° C water produces an outflow which increases from 0 to 4° C and then remains steady at 4° C until the interface reaches the bottom. The interface forms at the top when the mean temperature is 4° C. The time to empty the tank is 15/11 times the volume/discharge value because of the mixing.
Analytical equations are presented which define all of these effects.

ACKNOWLEDGMENTS

The tank and instrumentation were assembled by Mr. Ernesto Morala who also assisted in the testing program. His conscientious attention to accuracy has made the results quoted herein possible.

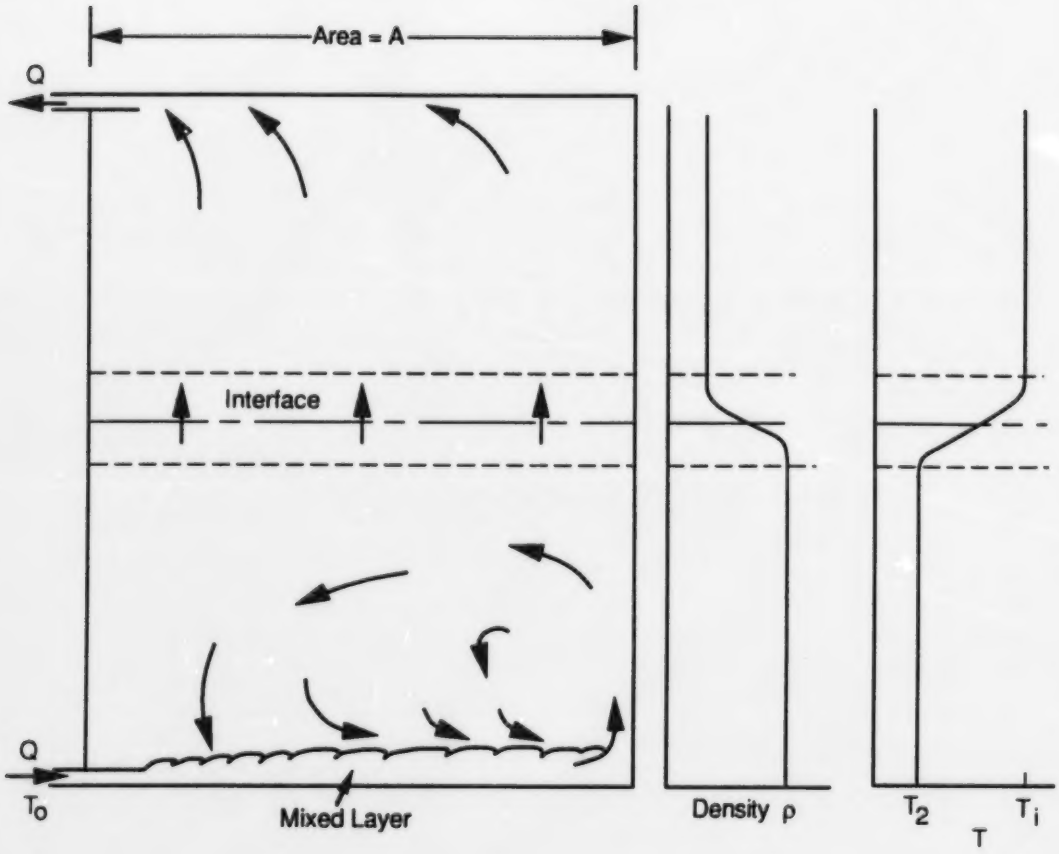

Figure 1 Definition sketch of flow and thermal regimes in a stratified thermal storage tank. Case of side inlet and exhaust is shown.

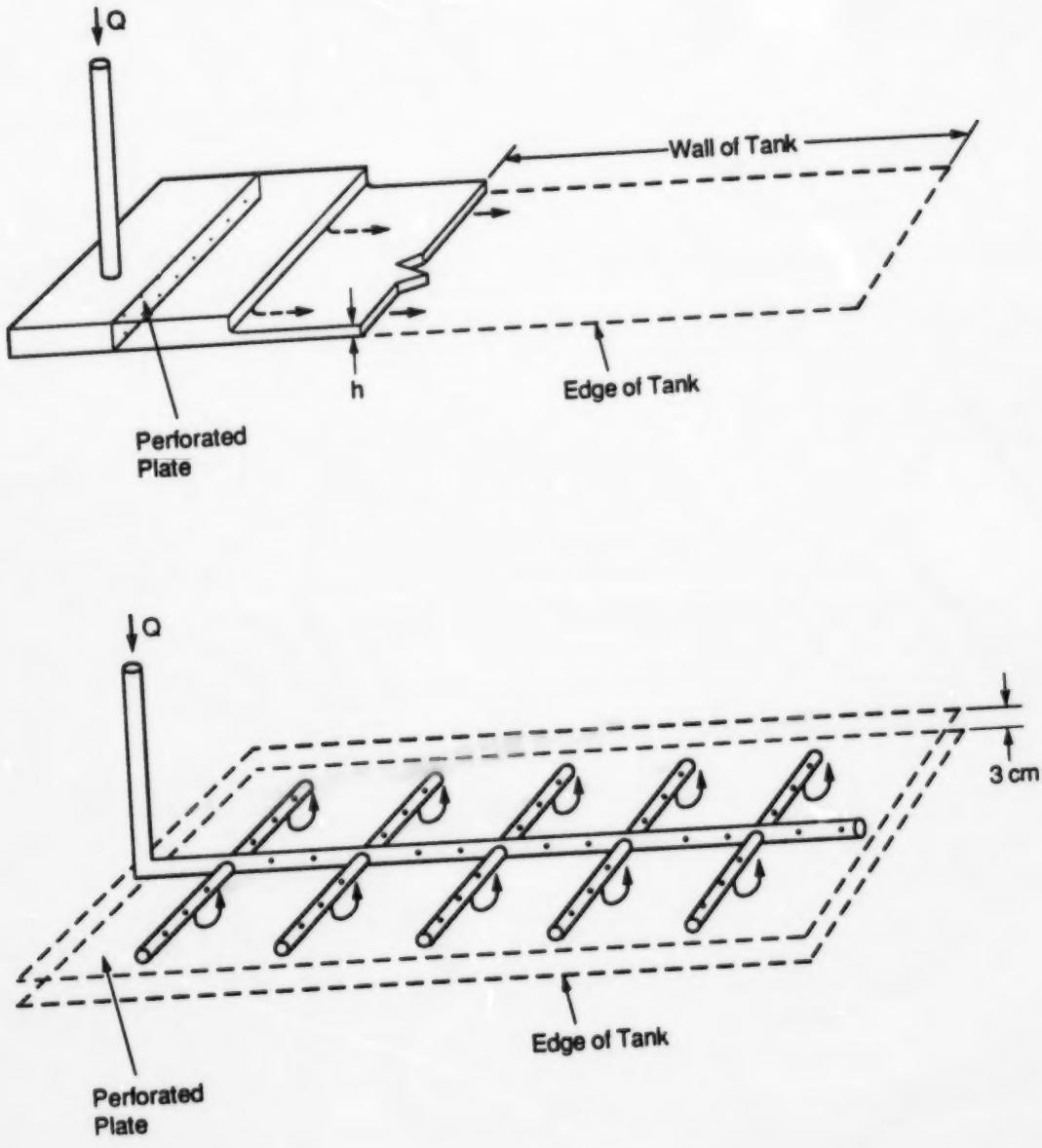

Figure 2 Sketch of side and bottom inlets used in tests

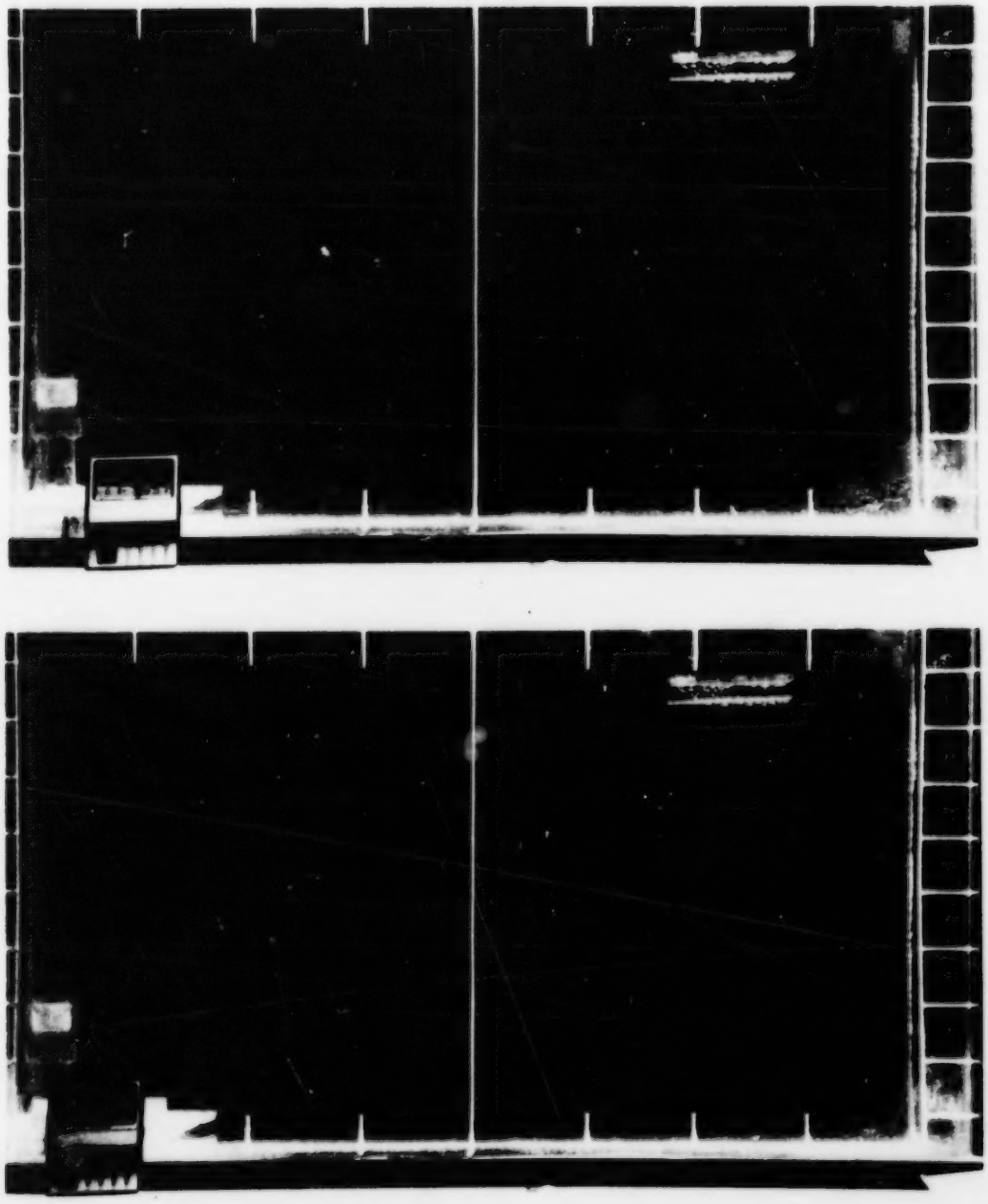

Figure 3a Photos of the flow pattern produced by a side inlet in Test #1. V = 2.65 cm/s, h = 2.2 cm, $\Delta\rho/\rho$ = 0.00185, Fr = 1.33

a. t = 120 s, Dye concentrated in mixing layer
b. t = 204 s, Interface established

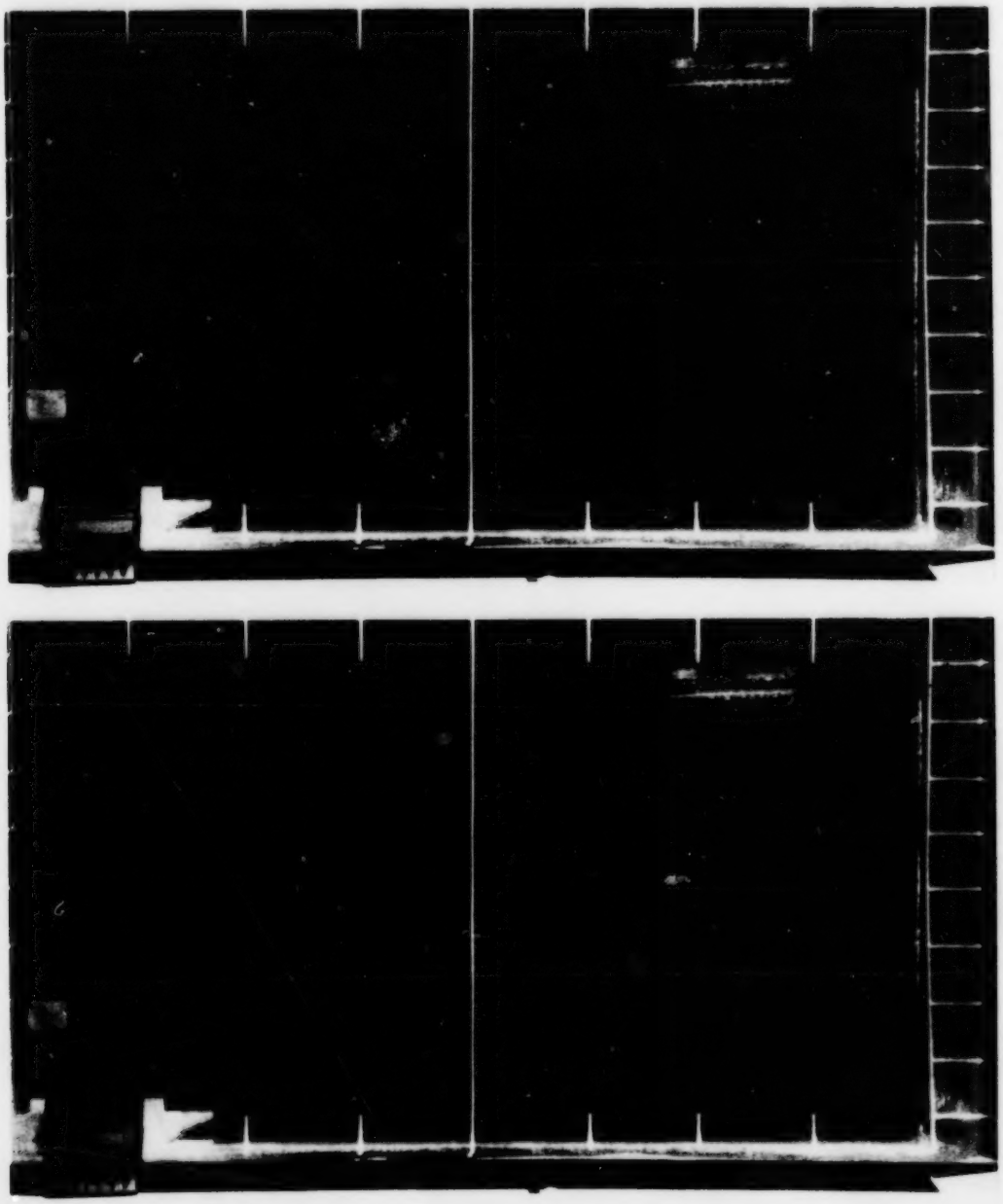

Figure 3b Photos of the flow pattern produced by a side inlet in Test #1. V = 2.65 cm/s, h = 2.2 cm, $\Delta\rho/\rho$ = 0.00185, Fr = 1.33

c. t = 276 s, Clear zone on right is a rapid recirculation region. Colored zone recirculates slowly.
d. t = 301 s

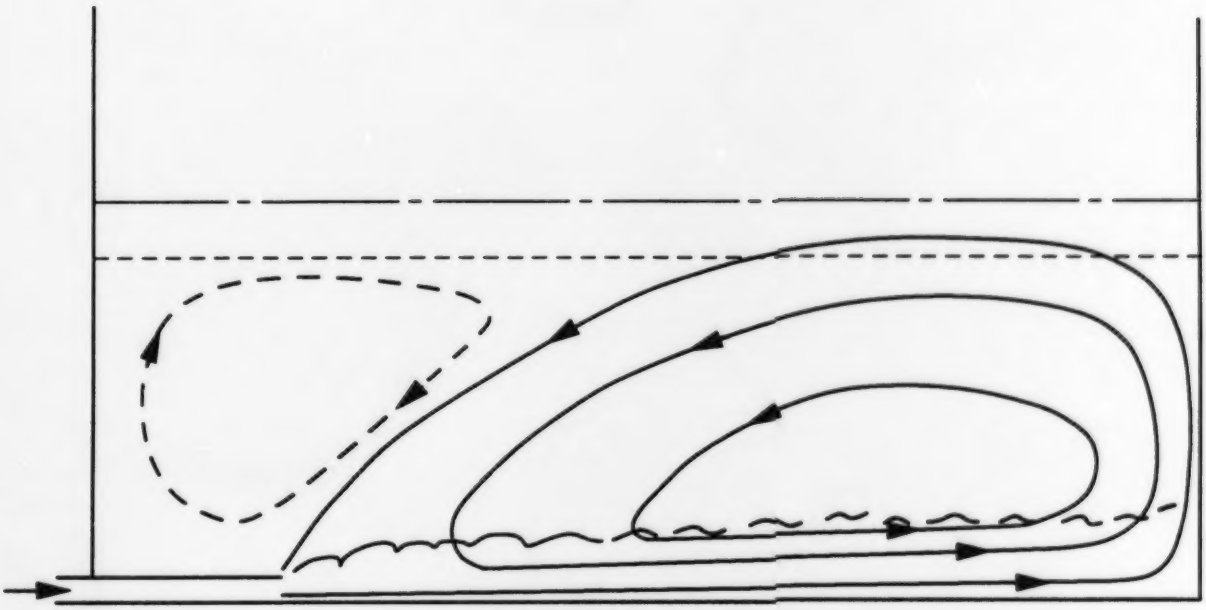

Figure 4 Sketch of streamline pattern deduced from the photos in Fig. 3.

APPENDIX A - SIMILARITY

Introduction

Studies of any flow phenomenon are meaningful only if the results can be applied to a larger scale manifestation of the phenomenon. Thus the transfer of results from the small tank used in these tests to the full size tanks is based on the concept of similarity. The fluid flow and temperature fields must have the same pattern. This is ensured if the ratios of forces and heat fluxes are the same in each case. Similarity can be applied to the entire tank, that is, it can be treated as a typical hydraulic model or it can be applied to the part of the tank where there is fluid motion. A second approach is possible for these tanks because there is a large volume which is stagnant and at uniform temperature. Its use makes possible a broader application of the results. This is discussed in more detail below.

Flow field

The dominant forces are inertia in the jet from the inlet and buoyancy due to the temperature differences. Flow is turbulent and mixing is driven by inertia. Viscous forces are small and do not contribute to either the pressures or mixing. The flow field is controlled by the ratio of inertia and buoyancy which is known as the Froude number

$$Fr = \frac{V}{\sqrt{g'L}} \qquad (1)$$

where V and L are a characteristic velocity and length
g' is the relative gravity = g $\Delta\rho/\rho$

The velocity and length to be specified depend on the interpretation of the problem. If the whole tank is being considered these are the mean upward velocity and height of the tank. On the other hand if the inlet and its effects are the subject the velocity is the jet velocity and the length is an inlet dimension. The practice now in tank design as discussed by Wildin(1990) is to define L as the height of the inlet opening h. This is based on the most common and cheapest inlet design now in use. This diffuser inlet consists of a pipe issuing into the area between a plate and the bottom (or top) of the tank. It may be a circular plate centred in the middle of the cross-section or a rectangular plate which spans one dimension of the cross-section. The recommended design value for this Froude number is unity or less.

One problem encountered with this definition of Froude number arises in the comparison of different inlet designs. Another typical design is an array of circular holes in a pipes which spread over the horizontal cross-section. The individual jets may be horizontal or, more effectively, vertical directed toward the tank bottom. For this L could be defined as the diameter of the holes or the diameter of the supply pipe. These are not really comparable to the slot height because the water does not flow at a constant depth from the outlet. It is preferable to use a different definition for the Froude number so that inlet design can be compared directly. This should be based primarily on the tank design parameters. One such definition is the flux Froude number, that is, one defined from the flows of volume, momentum and buoyancy into the tank. If a is the total area of the inlet openings then the volume flux or discharge is

$$Q = Va \tag{2}$$

the momentum flux per mass is

$$M = V^2 a \tag{3}$$

and the buoyancy flux is

$$F = g'Va \tag{4}$$

These three quantities can be combined into a dimensionless ratio in which the inertia flux is the numerator and the buoyancy flux is in the denominator. The result is the flux Froude number

$$Fr_f = \frac{M^{5/4}}{Q\,F^{1/2}} \tag{5}$$

Inserting Eqs.(3) and (4) gives this in terms of design parameters as

$$Fr_f = \frac{V^{5/4}}{g'^{1/2}Q^{1/4}} \tag{6}$$

In any installation the buoyancy g' is defined by the water temperatures and Q is the volume of the tank divided by the filling time. These are fixed by other factors and only V can be varied by the designer. Equation (2) shows that it is inversely proportional to a , that is

$$Fr_f = \frac{Q}{g'^{1/2}a^{5/4}} \tag{7}$$

This relation demonstrates that for a given inlet configuration the only way to reduce mixing is to increase the inlet area. It also defines the condition for similarity between lab and full scale tanks. The flow field will be the same if Fr_f is the same.

There is no restriction on g' evident in Eq(7) but one is imposed by the temperature field. Density is a function of temperature with a maximum at 4° C. Water is the only fluid which displays this variation so it must be used in the model at the same temperatures as the full scale tank. Thus g' is fixed and the Froude number can be maintained constant by changing Q and a. For example, if the lab scale tank is considered to be 1/10 the size of a full scale tank then the discharge must be $(1/10)^{2/5}$ of that in full scale.

References

Wildin, M.W., (1990), Diffuser design for naturally stratified thermal storage, Trans. ASHRAE,96(1), AT 90-13-4

APPENDIX B - DESCRIPTION OF THE EXPERIMENTS

All of the tests were conducted in a glass-walled tank 88.7 cm long 27.6 cm wide and 80 cm deep. The depth of water was 66 cm in all tests and the inflow and outflow discharges were metered continuously with drag type flow meters. The accuracy of these meters was about 3%.

Salt water and fresh water were used in preliminary tests to study the formation and motion of the interface. Dye was added to the salt water which was injected through the side inlet sketched on Fig. 2. Photos were taken periodically and the flow pattern below the interface was deduced. Measurements of the density profile confirmed it and formed the basis of an analysis of the profile.

For all of the tests using cold and warm water the tank was insulated by installing 5 cm slabs of foam insulation inside. This reduced the cross-section to 78.7 cm by 17.6 cm but prevented the transport of heat along the glass walls from the warm to cold water. This source of heat gain by the cold water was more important than the gain from the environment which was also reduced by the foam slabs. A thin polyethylene sheet was installed inside the insulation to prevent the water penetration and consequent increase of the conductivity of the slabs.

A rack of 55 thermocouples spaced 1 cm apart was installed vertically in the center of the cross-section. These were connected to an amplifier and a digital display reading to 0.1° C. All readings were taken by observing and recording manually. It is appreciated that a computer-based data acquisition system could have been used to save time in the recording and analysis but there was not sufficient time available to develop the system.

Figure 2 is a sketch of the two types of inlet used in the tests. The side inlet is a representation of the vertical slot opening most commonly employed in practice. Three opening heights were used. Upstream of the opening there was a duct 25 mm high which was the full width of the tank and there was a perforated plate across the upstream end of this duct. These items ensured that the flow from the inlet pipe was uniformly spread across the width and height of the opening. The second inlet design was a grid of pipes with 40 holes drilled in the bottom surface. The water issued downward and impinged on the bottom of the tank. Immediately above the grid a perforated plate of solidity 0.73 was installed horizontally. This design was intended to produce a uniform upwardly directed flow as in the bottom inlet used in practice. The distance from the plate to the tank bottom was 6 cm. In both installations there was an identical unit at the free surface of the top of the tank. This served as the outlet when cold water was replacing warm water and as the inlet when the flow was reversed.

APPENDIX C - PRELIMINARY TESTS

These were focused on the side inlet and were pursued to ensure that the inlet was operating as would be expected from previous studies of tanks. Several Froude numbers were used with the opening h = 22 mm but only one case will be presented here. The pattern and density profile were the same in each case. The development of the interface and the particle paths are illustrated in the photographs of Fig. 3. The first one was taken just after the salt water flow was started. Dye had been introduced into the inlet for the first few seconds of the test and it can be seen to be distributed uniformly though a mixing layer. This is a turbulent zone produced by the shear between the high velocity inflow jet and the stagnant water in the tank. The thickness of the mixing layer is approximately h. The three succeeding photos show a distinctive dye pattern. There is an intense line of color which is at the top interface. This line was the top of the mixing layer at the start. Measurements of its position revealed that it was moving up at a velocity Q/A which means that there is no flow across it. Below the interface there is a clear zone with a curved edge on the inlet side. Particles could be seen to travel across the bottom of the tank, rise up the opposite side wall and then return in a curved line to be entrained by the jet. These define the streamline pattern on Fig. 4. The curved line separates a zone on the left in which the particle motions were very small. There was a slow inflow to this zone at the inlet which means that the density in this zone is less than that in the larger zone to the right. When the inlet flow was stopped the curved line disappeared as the lighter fluid flowed over the heavier and blended into the interface. Figure 5 is a plot of the density profile measured at this time. The steep gradient at the top shows that the interface is very thin. One reason for this is the molecular diffusivity of salt in water which is about 1/1000 of the diffusivity of heat in water. The interface thickness is proportional to the square root of the diffusivity and this fact can be used as the basis of a theory to predict the thickness for a thermal interface. It can also be used to predict the interface thickness in a full-scale tank. A second reason is that the interface has been eroded from the bottom by the flow plotted on Fig. 4. The density of the inflow from the source is about 9% larger than the density of the layer below the interface which shows that the outflow from the jet is being continuously diluted by lighter fluid which must come from above the interface.

The flow pattern is the same as the one noted by Baines et al(1983) and the observed thin interface is consistent with those observed in similar studies. It is concluded that the side inlet is operating consistent with those in practice.

Reference

Baines, W.D., Martin, W.W. and Smith, D.M.,(1983), Development of stratification in a rectangular tank by horizontal inflow, J. Fluids Engin.,ASME, 105, 59-64.

APPENDIX D - DEVELOPMENT TESTS

These were a continuation of the preliminary tests but with warm (20° C) and cold (4° C) water. The objective was the understanding of the flow resulting from the buoyancy produced by temperature. The side inlet was used in every test. The tank was insulated so viewing of the flow pattern was not possible and the observations were temperature profiles. Sufficient information for analysis could be obtained from plots of a few profiles and the position of the interface center with time. Figures 6,7 and 8 present results for test #11 which was typical. The tank was initially filled with water at 22.1° C and the inflow was started at time zero at a discharge of 480 cm^3/min. The temperature of the water in the reservoir was 3.7° C but the inflow to the tank did not reach this value until 10 minutes has passed. The lines were full of the warmer water which had to be diluted and flushed into the tank. When the inlet flow was coldest the Froude number Fr = 1.33 and the flux Froude number Fr_f = 0.52 were typical of values in operating tanks. Figure 6 is a plot of the rise of the center of the interface which is defined by the mean temperature 0.5(22.1+4.2) = 13.2° C where 4.2° C is the temperature of the water at the bottom. The position of the center is defined solely by the conservation of the enthalpy introduced at the inlet. Thus it should be

$$\frac{dz_i}{dt} = \frac{Q}{A}\frac{T_i - T_0}{T_i - T_2} \qquad (8)$$

where T_2 = temperature of the water at the bottom of the tank.
For this test the slope of the points on Fig. 6 should have been 0.365 cm/min but the measured value was 0.322 cm/min. This indicates that there has been a loss or storage of enthalpy. The foam insulation has prevented heat flux from the environment but any water in the insulation would require cooling as the interface rises. An estimate of this effect can be made using the analysis for an infinite heat transfer coefficient between the tank and wall. If the wall has the specific heat of water this gives a correction factor for dz/dt which is

$$1/(1 + A_w/A)$$

where A_w is the cross-section area of the wall. The total wall area gives this factor as 0.74 whereas the data gives a factor of 0.88. This indicates that the heat capacity of the wall is about 1/5 of that of a layer of water of the same thickness. The measured factor was the same in all experiments with warm and cold water.

Temperature profiles should naturally be compared with those which would be produced by heat conduction from the warm to cold water. If two materials are brought into contact at time zero then the temperature at a later time is given by the solution quoted by Carslaw and Jaeger (1959)

$$T = \frac{1}{2}(T_1 + T_2) + (T_1 - T_2)\, erf\left(\frac{\Delta z}{2\sqrt{\alpha t}}\right) \qquad (9)$$

where α = thermal diffusivity = 1.41×10^{-3} cm^2/s (water at 4° C).
Mixing of the water at the jet outlet produces an interface with an initial temperature profile. Thus at any later time the interface should be thicker than that given by Eq.(9). Figure 7 is a plot of

the measured profile taken 12 minutes after the cold water reached the inlet . This is actually 20 minutes after the pump was started. The profile which has been matched at the center is close to an error function which demonstrates that the effect of mixing is similar to that of conduction. The magnitude of the slope defines the effective time origin of the profile which is defined by differentiating Eq.(9)

$$\frac{dT}{dz}_{z=0} = \frac{1}{2}\frac{(T_1 - T_2)}{\sqrt{\pi \alpha t}} \tag{10}$$

Equivalently an interface thickness δ can be defined from the slope by

$$\delta = (T_1 - T_2)/\frac{dT}{dz}_{z=0} = 2\sqrt{\pi \alpha t} \tag{11}$$

The profile plotted on Fig. 7 has a slope 4.1° C/cm at the center which, from Eq.(10) indicates a time of 17 minutes since origin. The actual time of 12 minutes indicates that the mixing process is equivalent to 5 minutes of time with conduction. As time progresses the slope should decrease and δ should increase following these equations. It is apparent from Fig.8 that this is indeed the trend. However, the measured slope of 1.4° C/cm at the center corresponds to an elapsed time of 113 minutes while the actual elapsed time is 117 minutes. This gives a time of -4 minutes at the start of the test. The change from +5 min at 20 min to -4 min at 117 min is due to the circulation pattern seen in the photos. The flow along the interface erodes the slightly warmer water and brings it back toward the mixing zone at the bottom. Thus conduction increases the thickness and recirculation reduces it. The magnitude of this reduction cannot be predicted by elementary theory but this test has provided a measure of the effect. At 12 minutes δ = 4.35 cm and at 117 minutes δ = 12.1 cm whereas conduction would have produced δ = 13.6 cm. The mixing of this warm water carried by this circulation with the inlet flow produces the temperature at the bottom of 4.2° C which is warmer than the inflow temperature of 3.7° C. These results demonstrate that the region below the interface is not at rest as has been assumed in some previous work at these small Froude numbers.

APPENDIX E - TESTS WITH SIDE INLET

The test discussed above was repeated using inflow temperatures which were less than 4° C. Effects of the density maximum are clearly seen by comparing the results to those in section 5. Most important is the temperature below the interface which is very close to 4° C for any inlet temperature. This means that, following Eq.(8), the rate of rise of the interface center as plotted on Fig. 9 is faster than for the warmer inlet temperature. The measured slope is 0.91 times the slope given by Eq.(8) as would be expected from the discussion above. The temperature profile plotted on Fig.10 is close to the error function in the zone above the center but deviates appreciably in the lower part. This shows that there was intensive turbulent mixing which eroded the interface. Further evidence is given by the effect elapsed time calculated from the center temperature gradient. This is 58 minutes compared to the actual 100 minutes which had elapsed since the cold water started flowing into the tank. Thus a second mechanism has been identified which erodes the interface. The cold but light masses of water leaving the inlet develop a vertical velocity by buoyancy. These are turbulent and impact the interface at an elevation where the temperature is above 4° C. These are brought to rest and the turbulence causes rapid mixing with the surrounding water. The rising masses can be seen in the zig-zag temperature profile plotted on Fig.11. In all of the zone below the interface the profile shows spatial oscillations.

Test #14 used an inlet jet thickness of 8 mm for which $Fr_t = 0.22$. The same test was also conducted with a 4 mm ($Fr_t = 0.52$) and 22 mm ($Fr_t = 0.062$) thicknesses. The measured interface rise rates and temperature profiles were the same as those plotted on Figs.9 and 10. It is concluded that the dominant mechanism of interface development in this case is the buoyancy driven erosion.

APPENDIX F - BOTTOM INLET TESTS

The recirculation pattern seen in the photos is obviously driven by the horizontal momentum of the inlet jet so it could be eliminated by introducing the flow vertically. This is the concept behind the bottom inlet design. In addition the vertical momentum is reduced by directing the inflow against the bottom of the tank. With this arrangement large turbulent eddies are created as the individual jets are deflected off the surface. A perforated plate with solidity 0.73 was installed 1 cm above the supply grid to break up these eddies. The definition of flux Froude number for this design depends on which of the flow areas is most effective. The total outlet area of the grid is 7.13 cm^2 which defines $Fr_f = 0.46$ but the total open area of the perforated plate is 374 cm^2 which defines $Fr_f = 0.003$. Thus if the plate produces a uniform, low turbulence upward flow one would expect that there would be no recirculation.

The measurements are summarized on the plots of Figs.11, 12 and 13. First it is evident from Fig.11 that the mean temperature of the zone below the interface is less than the 4.1° C of Test #14. It is 3.74° C which is a 0.36° C reduction. The interface rise on Fig.12 is less than the value given by Eq.(8) as noted for the previously discussed tests. The temperature profile shows the deviation from the error function profile mentioned above indicating that the buoyancy driven mixing is strong here too. There does not appear to be an effect of this mixing on the thickness of the interface. The temperature gradient at the center gives an effective elapsed time of 126 minutes which is very close to the actual time since introduction of the cold water.

In summary, these tests of the two types of inlet show that the advantage of the bottom inlet is a lower mean temperature and hence an increase in storage of cold. In both designs, the mean temperature is close to 4 C° and the interface consequently rises faster than the water particles.

APPENDIX G - HORIZONTAL SCREEN INSTALLED ABOVE INLET

Introduction

One feasible scheme for maintaining an unstable density profile uses horizontal screens. These could be either wire mesh gauze or perforated plates. In either case it is envisioned that the solidity is large enough for the pressure drop due to the upward flow during tank filling to balance the buoyant force due to the light fluid on top. Solidity is the ratio of solid area to the total area and it must be larger than 0.5 for structural strength. The magnitude of the solidity required to prevent the downward flow is deduced from the drag force produced by the imposed upward flow velocity V. This is expressed as a pressure drop coefficient k defined by

$$\Delta p = k\frac{\rho V^2}{2} \qquad (12)$$

A plot of measured and calculated values of k is given by Baines and Peterson(1951). The perforated plate used in these tests had solidity s = 0.73 for which k = 20.

If there is no upward flow, i.e. V = 0, then in an unstable density profile an upward flow of magnitude V' passes through half of the openings in the screen and a downward flow of equal magnitude passes through the other half. The buoyant force is equal to the inertia in the jets coming through the openings so

$$V' = \frac{(1-s)}{2}\sqrt{\frac{gd\Delta\rho}{\rho}} \qquad (13)$$

where d = effective diameter of the openings

If there is both an imposed upward flow V and an unstable density distribution then for

V < V' then the downward flow is reduced to V' - V.

V > V' there is no downward flow and the screen acts as a barrier to the heavier fluid.

Test of a single screen

A perforated plate of s = 0.73 was installed above the bottom inlet unit described above. The same inflow was used as in the previous tests which produced an upward flow V = 0.006 cm/s. Figure 14 is a plot of the progression of the center of the interface and the temperature profile is plotted on Fig. 15. Several features on these plots are significant:

- the temperature of 2.5° C below the screen is less than the temperature of the water just above it which is 3.9° C. This shows that the screen has inhibited the downward flow, the actual size of which can be estimated from conservation of heat. The heat balance for the layer between the screens when the conditions are steady states that the flux from below equals the flux from above, i.e.

$$0 = V''(T_2-T_0) - (V+V')(T_1-T_0)$$

Solving this for V" gives

$$V'' = V(T_2-T_0)/(T_2-T_1) = V(2.5-1.4)/(3.9-2.5) = 0.8\ V$$

for the inflow temperature T_0 = 1.4° C. The downward flow velocity is thus about 0.005 cm/s. The value given by Eq.(13) for this temperature difference is about 0.008 cm/s. This is reasonable agreement considering the uncertainty in Eq.(13) and the accuracy of temperature measurements.

- the mean slope of the centerline rise on Fig. 14 is equal to that on Fig. 12. This would be expected since the mixing means that slope is controlled by the heat balance in the layer below the interface. However, the slope is not constant. The interface rises in a series of fast and slow periods. This is probably due to a process of accumulation of lighter fluid just above the screen which rises suddenly when it reaches a critical size. There is no evidence of accumulation in the measured profiles, it probably occurs at a location far from the measuring section. The changes to the profiles seen in Fig. 15 are, however, consistent with this interpretation. Four measured profiles are shown in this Figure along with the error function profiles which would be produced by steady upward flow and pure conduction of heat. The error function has been fitted to the slope of the lowest profile so if conduction were the only mechanism the other measured profiles would be matched by the theoretical curves. It is significant that:

  - the center of the profile moves more slowly for the first twenty minutes and then catches up in the last ten minutes showing that there has been a large transport of cold water during this time.

  - the measured profiles do not change shape while the theoretical profiles become steeper because conduction increases the thickness. The turbulent mixing produced by the rising colder (and lighter) water tends to make the profile flatter, that is, to produce a step change in temperature.

  - the measured profiles deviate from the error function more below than above the centerline. This, too, indicates turbulent mixing.

Theory for mean temperature

As an example, consider a single screen which prevents downward flow and is installed at elevation h_1 in a tank of depth H. Define t_f as the time to fill the tank and t_1 as the time for the center of the interface to reach h_1. Thus since the temperature under the interface is 4° C.

$$t_1 = \frac{h_1}{H}\frac{(T_i-4)}{(T_i-T_0)} \qquad (14)$$

For no downward flow and thorough mixing below the screen the temperature here at t_1 is

Table I contains numerical values calculated for a range of h_1 with T_0 = 0° C and T_i = 15° C. The limit of H/h_1 = 1 is the case of no screen. The temperature of the cold layer below the screen

$$T_{1f} = (4-T_0)\exp(-(\frac{H}{h_1} - \frac{(T_i-4)}{(T_i-T_0)}) \quad (15)$$

decreases as screen elevation is lowered and there appears to be a real benefit to the installation at small h_1. However, the important property is the mean temperature T_m of all the water in the tank. The temperature of the water above h_1 must also be calculated. This is determined by first calculating the time for the center of the interface to reach the top of the tank and then assuming continuous mixing above h_1 with the upward flow through the screen. The equation for t_{2f} the time to reach H is defined implicitly by

$$(T_i-4) = (T_i-T_0)\frac{t_f}{t_f} - (4-T_0)\frac{h_1}{H}(1-\exp(\frac{t_2 H}{h_1 t_f} - \frac{(T_i-4)}{(T_i-T_0)})) \quad (16)$$

and the final temperature of the layer above the screen is

$$T_{2f} = (4-T_0)\exp(-\frac{Q(t_f-t_2)}{Ah_1}) \quad (17)$$

From these the mean temperature is

$$(T_m-T_0) = \frac{1}{H}(h_1(T_{1f}-T_0)+(H-h_1)(T_{2f}-T_0)) \quad (18)$$

Table I Temperatures produced by one screen

h_1/H	T_{1l} °C	T_{1m} °C
0		3.06
0.1	0	2.47
0.2	0.06	
0.3	0.14	
0.4	0.68	
0.5	1.13	
0.7	1.57	
0.8	2.39	
0.9	2.74	2.74
1.0	3.06	3.06

A few representative values have been calculated and listed in Table I. It shows that for the most favourable location there is a reduction of 0.5° C. This is a 5% increase in thermal capacity and may justify the cost of installation. Because of this improvement from one screen it is worthwhile examining the benefits of installing many screens. It should be noted that a 1° C or 10% increase in capacity exists for no screen. This is a consequence of inflow at 0° C compared to 4° C.

As a second example the temperature profile will be calculated for 9 screens installed 0.1 h apart from 0.1 H to 0.9 H. These effectively separate the tank into 10 layers the temperature within each one of which is uniform. This can be calculated by a simple heat balance for each layer. The rate of increase of enthalpy of the layer is equal to the difference between inflow of heat from the layer below and the outflow to the layer above. For the ith layer

$$\frac{d}{dt}(\Delta h_i T_i) = Q(T_{i-1} - T_i) \qquad (19)$$

This equation is solved first for i = 1 to give T_1 the result inserted in the equation for i = 2. The solution for this one gives the input for i = 3 and the sequence of solutions proceeds to i = 9. Some of the solutions are

$$T_i - T_1 = (T_i - T_0)\exp(-t') \qquad (20)$$

$$T_i - T_2 = (T_i - T_0)[1 + t']\exp(-t') \qquad (21)$$

where $t' = Ht/\Delta h t_r$.

$$T_i - T_3 = (T_i - T_0)[1 + t' + \frac{t'^2}{2}]\exp(-t') \quad (22)$$

$$T_i - T_4 = (T_i - T_0)[1 + t' + \frac{t'^2}{2} + \frac{t'^3}{3!}]\exp(-t') \quad (23)$$

Table II Temperature profile produced by 10 screens

I	T_{if} °C	I	T_{if} °C
1	0.0006	6	1.00656
2	0.0675	7	1.9515
3	0.0420	8	3.3030
4	0.1545	9	4.9920
5	0.4395	10	7.3995

It is readily noted that the expression in the square bracket adds a term for each increase in i which is of magnitude $t'^{(i-1)}/(i-1)!$ so the result for any value of i can be directly stated. In the one case which has been calculated $\Delta h = 0.1$, $T_i = 15°$ C and $T_0 = 0°$ C. Table II is a listing of the final temperatures of each layer.

The mean temperature is the mean of T_{if} which is 1.8540° C. This represents an increase of 21% in capacity over a tank at 4° C. An estimate of the cost of installing this large number of screens indicates that this would be large. It is concluded that there is not a financial benefit to this installation.

References

Baines, W.D. and Peterson, E.G., (1951), An investigation of flow through screens, Trans. ASME, 73, 451-480

APPENDIX H - VERY COLD WATER INLET AT TOP

It is evident from the discussion above that the tank fills with water at 4° C if the inlet temperature is 0° C. The time for the interface to reach the top is less than the filling time since enthalpy is conserved. Thus in the interval between these two times the tank can be supplied with water at 0° C and the mean temperature reduced steadily. In the foregoing section it was calculated that the temperature would be 3.06° C if the bottom inlet were employed and the tank mixed by the buoyancy driven motion. In this section the consequences of reversing the flow and using the top grid as the inlet will be considered. This reversal is employed when the tank is discharged, that is, filled with warm water. The inflow is lighter at 0° C than the tank water so an interface should form with little mixing of the 4° C water.

Figure 16 is a plot of the temperature profiles measured in Test #24 which simulated these conditions. The inflow was 920 cm^3/min at 1.6° C and the initial temperature was 4.5° C. This defined a flux Froude number of 4.9 based on the grid outlet area. This is relatively large and some mixing would be anticipated. This is evident in the thickness of the interface and in its position. The center moved at a rate 1.4 times the value given by Eq.(8). This advance of the interface at a speed faster than the water particles means that it is being eroded by the inertia of the inlet flow.

The accuracy of the profiles is not sufficient to permit an evaluation of the effective time since the start but it is likely to be longer than the elapsed time because of the erosion. One evident feature is the fact that the profiles coincide near the top and bottom. The area between the curves is the total enthalpy form the inlet in the intervening 9 minutes.

The fact that the water at the bottom remains at 4° C in the design case permits the determination of the mean temperature at 2.93° C. This is a gain of 0.13° C over the case of using the bottom inlet.

APPENDIX I - RECHARGING THROUGH TOP INLET.

A question which has not yet been addressed is the temperature regime which will be produced by the refilling with 15° C water of a tank of 0° C water. Will the mixing produce large scale motions that will lead to masses of water at an intermediate temperature. The coldest possible temperature is desired for the operation of cooling units. This was investigated by filling the tank with 0.9° C water which was the coldest available and introducing 16° C water through the top inlet. These define T_i and T_0 respectively. Figure 16 is a plot of three temperature profiles measured during the run. The one on the left was measured before the run. For the first 15 minutes the temperature increased almost uniformly over the depth. At most there was a difference of 1.5° C between top and bottom. When a layer of 4° C water had been established at the top, an interface developed and was fully formed at 25 minutes as seen in the center profile. At 65 minutes the interface has descended and the layer below it is all at 4 C. In the elapsed time of 40 minutes the center moved 13.7 cm. Equation (8) gives a tine of 13.9 cm and the close agreement shows that there is no erosion of the interface.

These results show that the design of a tank can be based on the following principles:

- the tank mean temperature increases uniformly until sufficient heat has been added to bring it from the initial temperature to 4° C.

- following this an interface forms in the same way as for the introduction of cold water into the bottom of a warm tank.

A question arises at this point about the outflow temperature which would be produced if the tank had been recharged with a layer of ice. This is a scheme now used to increase the capacity of existing small tanks and one that has not been tested widely. Introducing ice into the top of the tank obviously reduces the mean temperature. However, if the mean temperature is above 4° C this is not an efficient process. As can be seen from the experiments discussed above the heavier cold water would sink to the bottom and cause intensive mixing. On the other hand, for a mean temperature > 4° C all water melted from the ice would be lighter and would stay at the top. This principle was verified in Test #25 in which the tank was filled with water at 0.8° C and a layer of ice chips was placed over the horizontal screen. The inflow of 16° C water was directed upward, impinged against the free surface and then flowed through the 2 cm layer of ice. A weak temperature gradient between the top and bottom was established and the mean temperature increased with time from 0.8 to 3.8° C. It stayed constant until all the ice melted which indicated that the temperature of the outflow is about 4° C which is the same result as in a tank initially at 0° C.

APPENDIX J - DENSITY OF WATER

The following table has been prepared from the formula describing the density of water in the range from 0° C to 100° C. Density is quoted in kg/m^3.

$$\rho = (999.8396 + 18.224944\,T - 0.0079222\,T^2 - 5.54486\times10^{-5}\,T^3 + 1.497562\times10^{-7}\,T^4 - 3.932952\times10^{-10}\,T^5)/(1 + 0.01859725\,T)$$

T °C	T °F	ρ kg/m^3	$\Delta\rho/\rho$
0	32	999.8396	1.325×10^{-4}
0.5	32.9	999.8714	1.071×10^{-4}
1	33.8	999.8987	7.342×10^{-5}
1.5	34.7	999.9216	5.048×10^{-5}
2	35.6	999.9400	3.211×10^{-5}
2.5	36.5	999.9544	1.776×10^{-5}
3	37.4	999.9645	6.627×10^{-6}
3.5	38.3	999.9702	1.183×10^{-6}
4	39.2	999.9743	1.831×10^{-7}
5	41.0	999.9642	7.935×10^{-6}
6	42.8	999.9404	3.168×10^{-5}
10	50	999.7000	2.721×10^{-4}
15	59	999.1006	8.715×10^{-4}
20	68	998.2059	1.766×10^{-3}
25	77	997.0475	2.925×10^{-3}

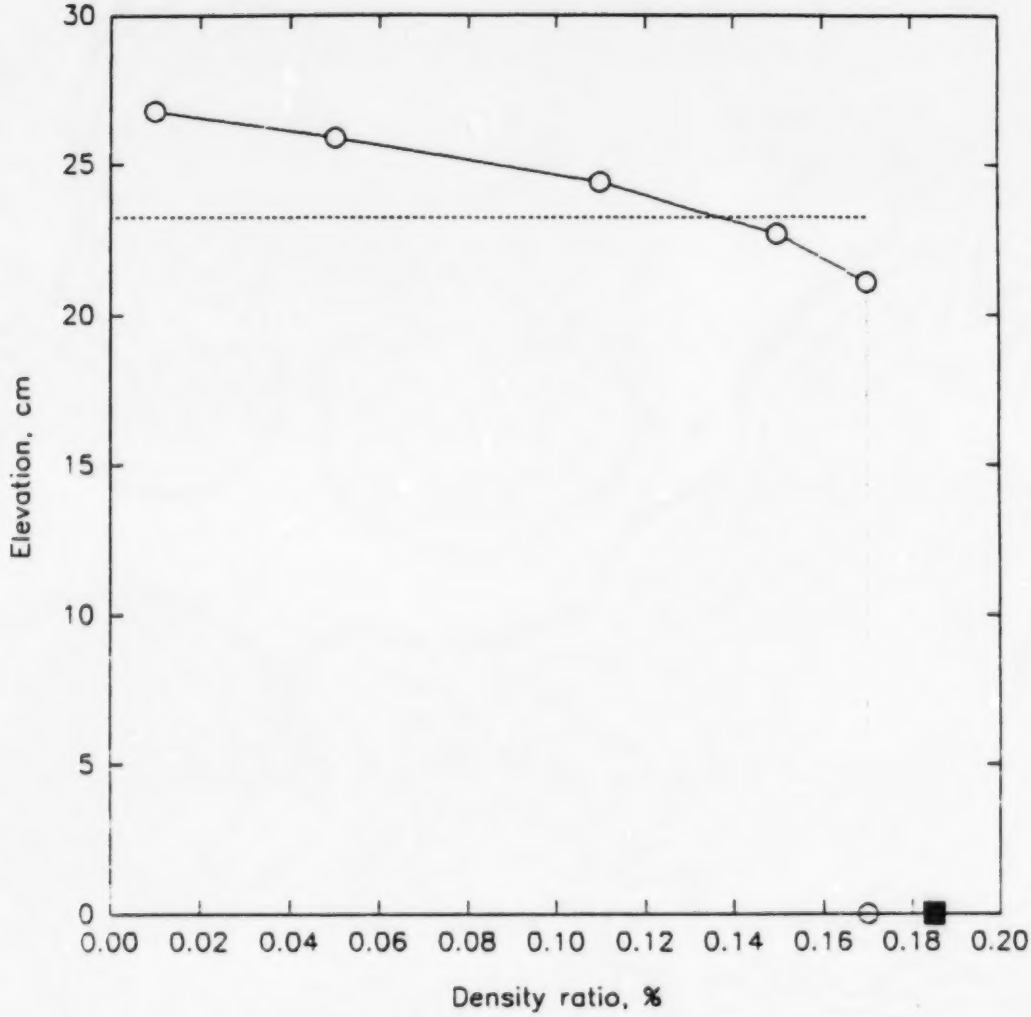

Figure 5 **Density profile measured at t = 510 s in a Test #1. Density ratio = $\Delta\rho/\rho$. Open circles are measurements. Solid square is measured value at inlet. Dotted line is elevation of water moving at velocity Q/A. Mean density is at an elevation 1.09 higher which shows effect of dilution by recirculation.**

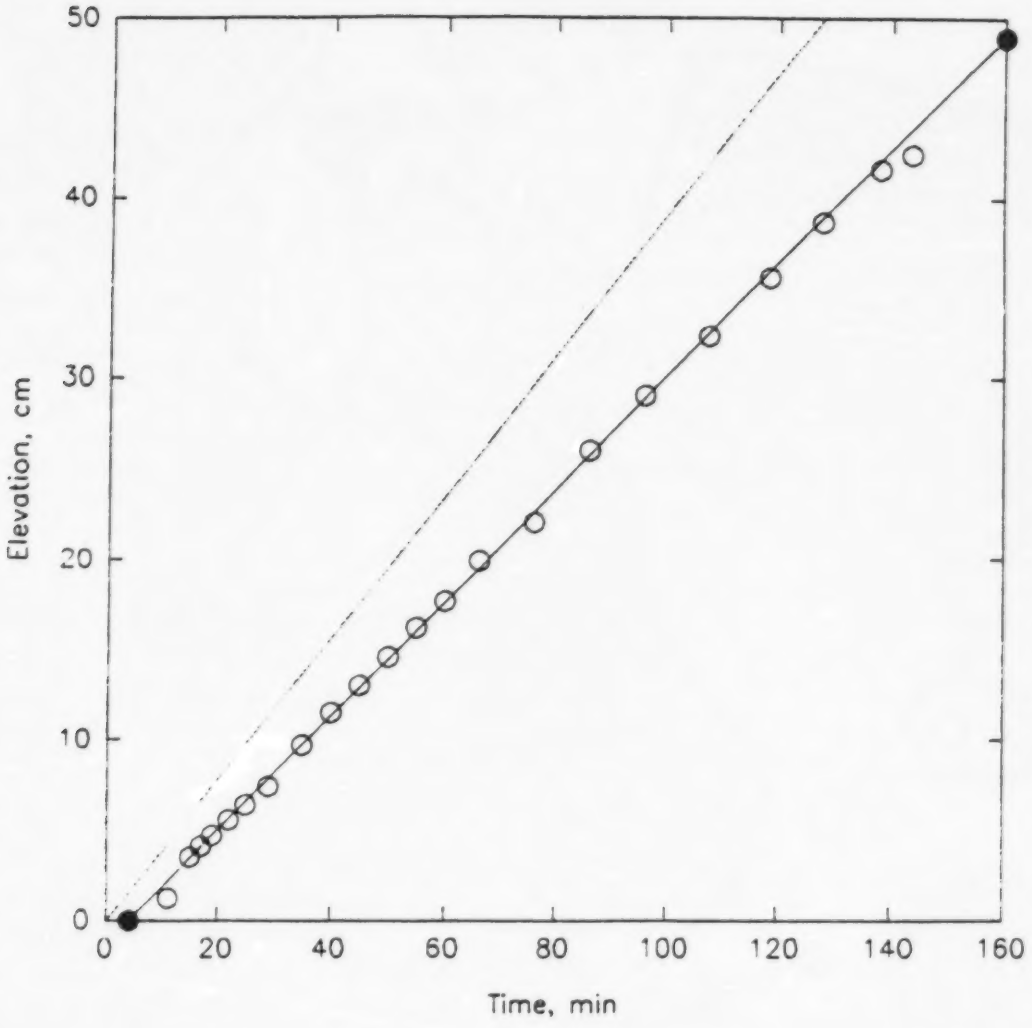

Figure 6 Elevation of the center of the interface as function of time in Test #11. In this test initial tank temperature T_i = 21.3° C, the inflow temperature T_0 = 4.4° C, V = 0.12 cm/s, h = 0.37 cm. Solid line is fitted to measurements. Dashed line is Eq.(8).

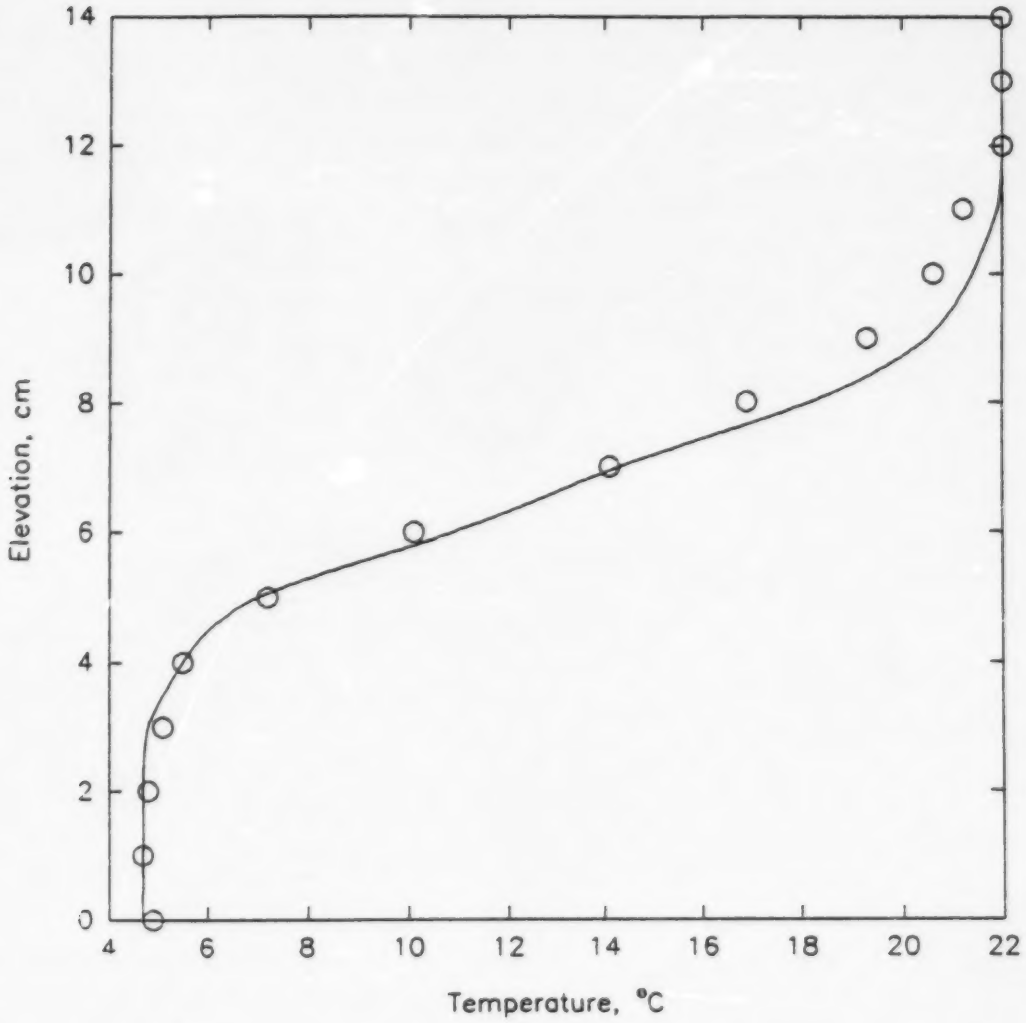

Figure 7 Temperature profile measured at t = 12 min in Test #11. Solid line is error function fitted to temperature gradient at center of interface.

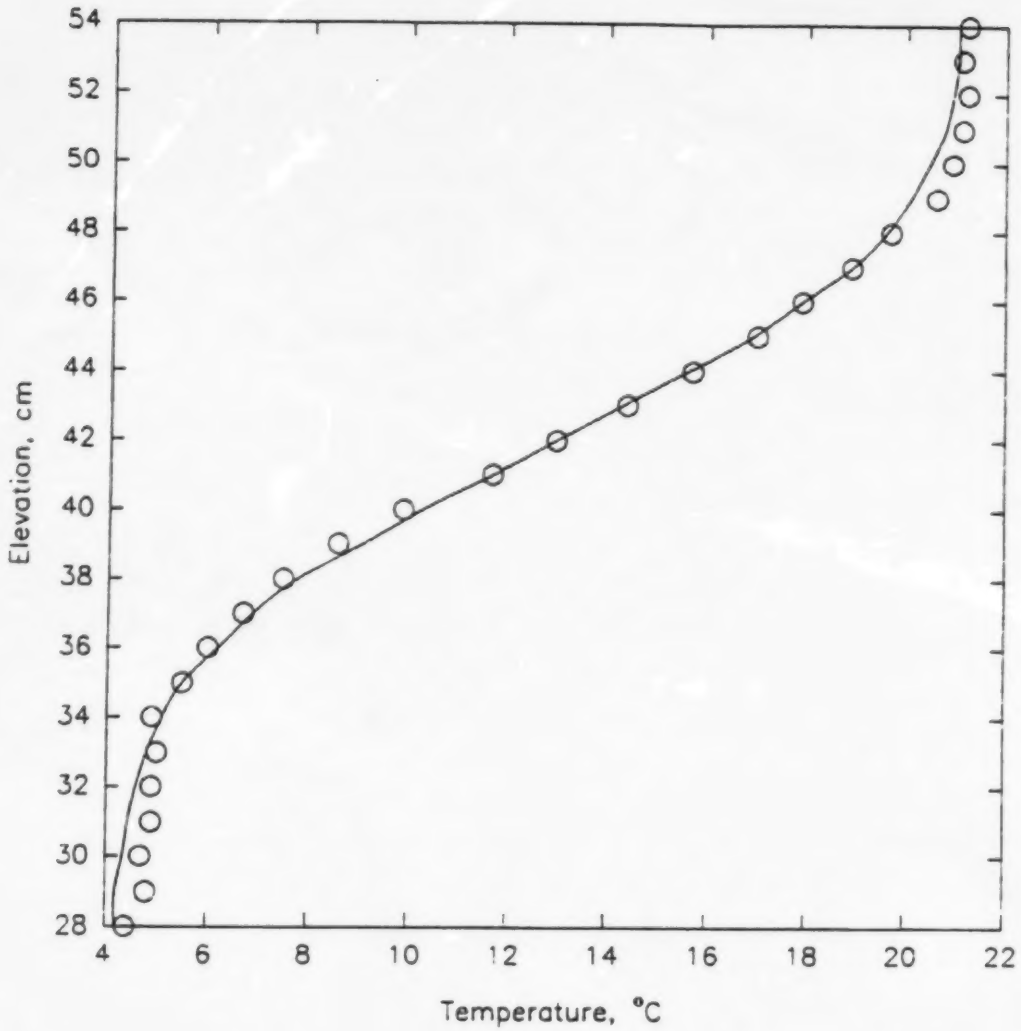

Figure 8 Temperature profile measured at t = 117 min in Test #11. Solid line is error function fitted to temperature gradient at center of interface.

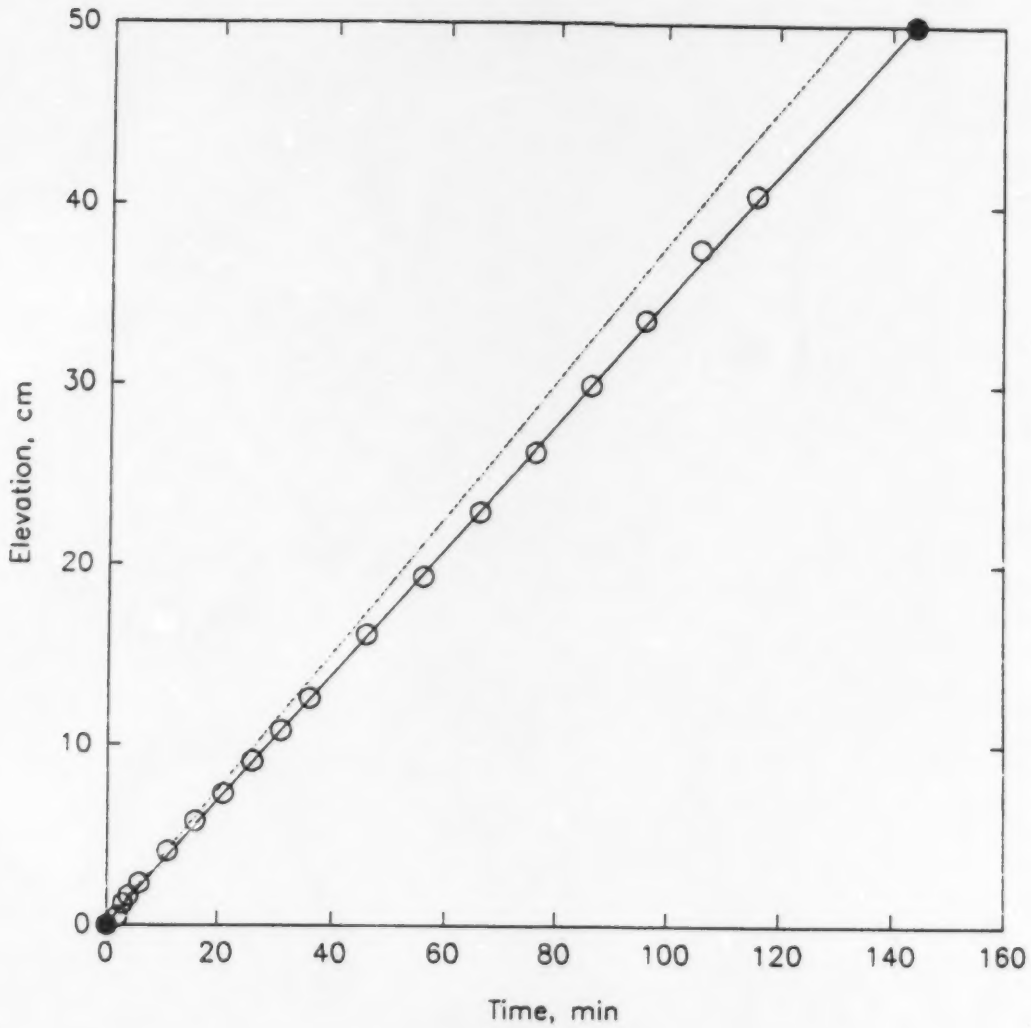

Figure 9 Rise of center of interface in Test #14. T_i = 20.2° C, T_0 = 2.4° C, V = 0.061 cm/s, h = 0.74 cm. Solid line is fitted to measurements, dashed line is Eq.(8).

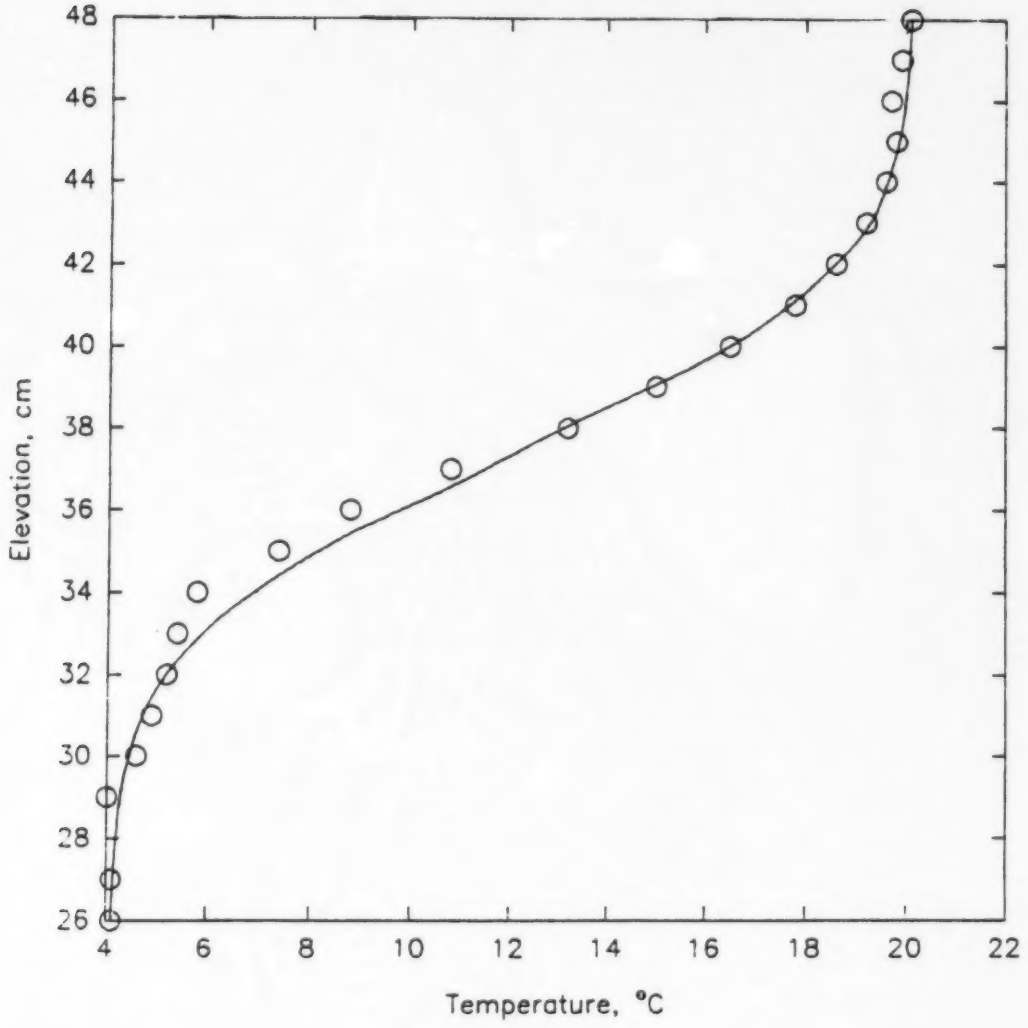

Figure 10 Temperature profile measured at t = 100 min in Test #14. Solid line is error function fitted to temperature gradient at center of interface.

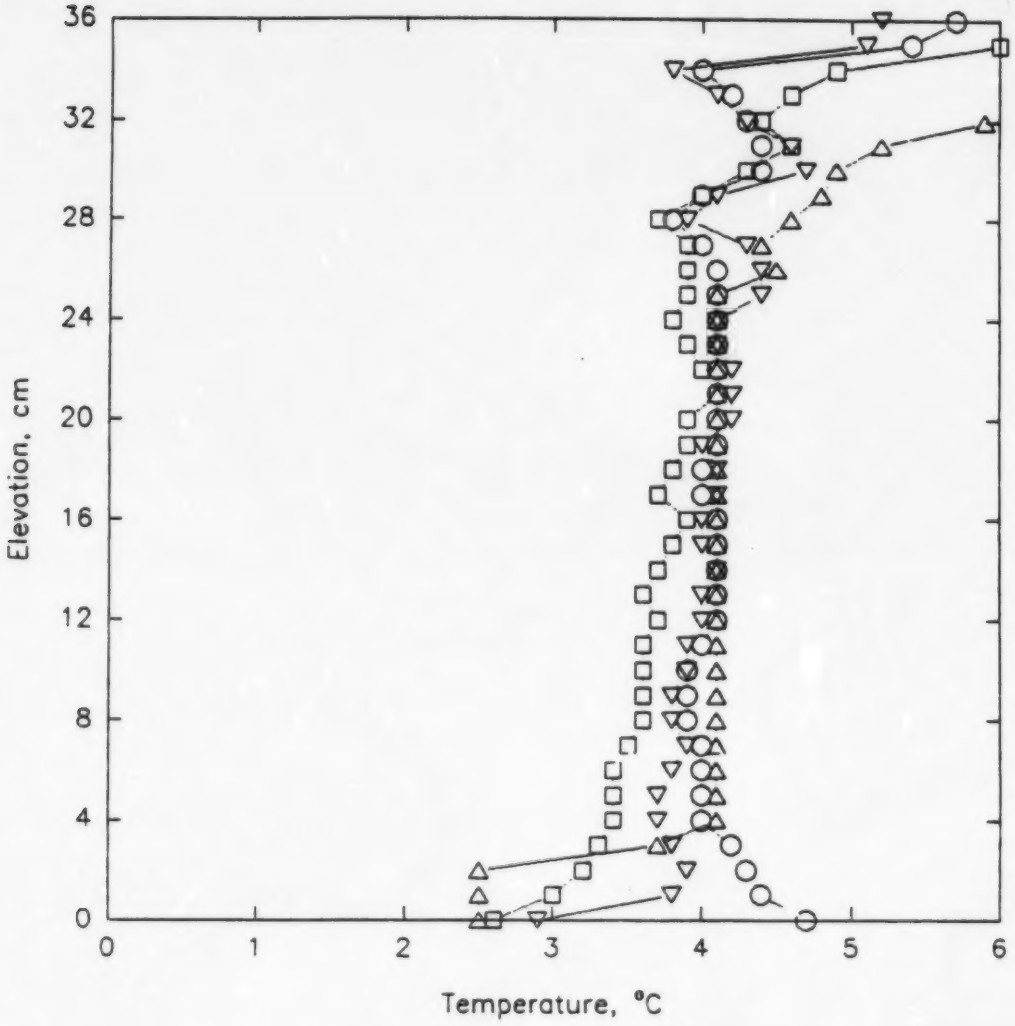

Figure 11 Temperature profiles in the region below the interface for four tests with inlet temperatures less than 4 C. Open circles are Test #14. Triangles down are Test #16 which was identical to Test #14 except that a horizontal perforated plate was placed above side inlet. Open squares are Test #18 which used the bottom inlet and horizontal screen. Triangles up are Test #21 which employed a second screen 3 cm above the bottom inlet. Temperature below screen is close inlet temperature but temperature above screen is uniform.

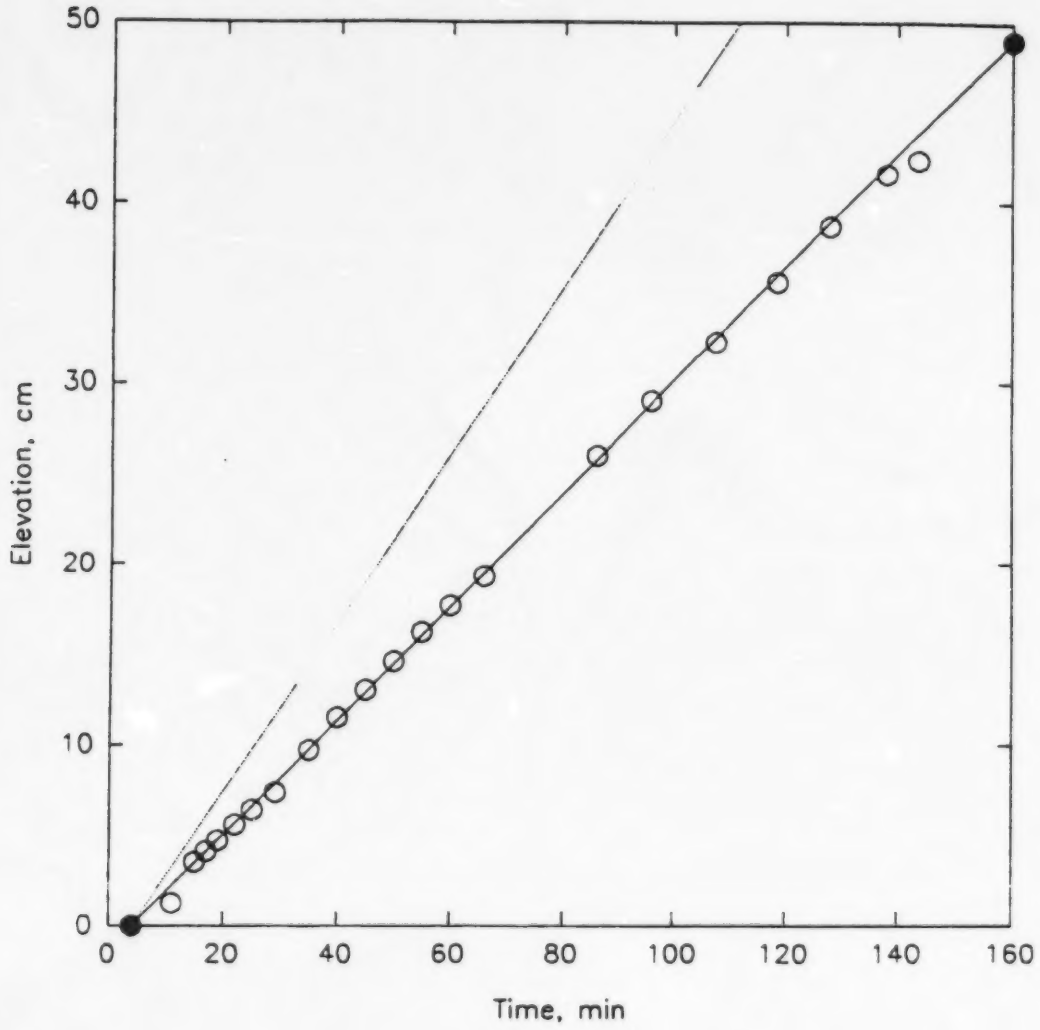

Figure 12 **Rise of center of interface in Test #18. Bottom inlet employed. T_i = 19.7° C, T_o = 2.3° C, V = 1.12 cm/s, a = 7.12 cm². Solid line is fitted to measurements, dashed line is Eq.(8).**

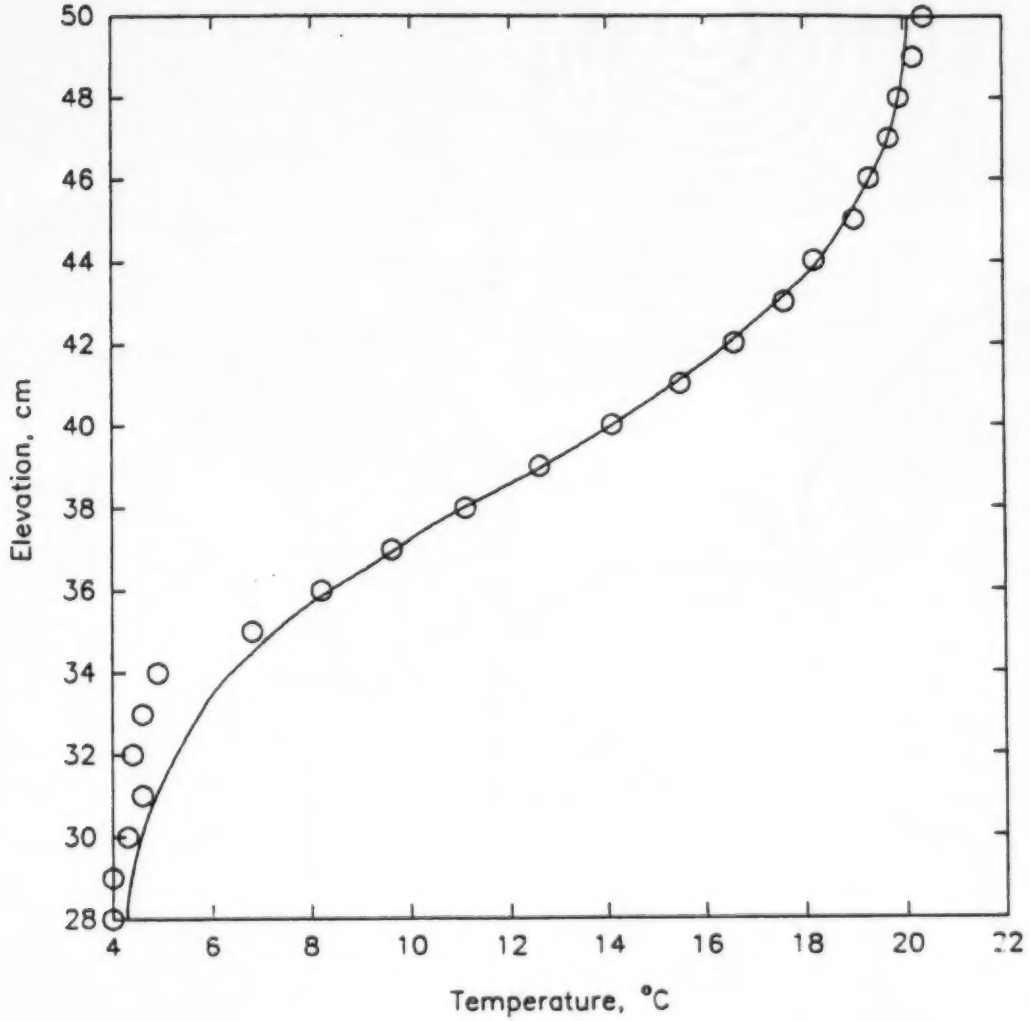

Figure 13 Temperature profile measured at t = 125 min in Test #18. Solid line is error function fitted to temperature gradient at center of interface.

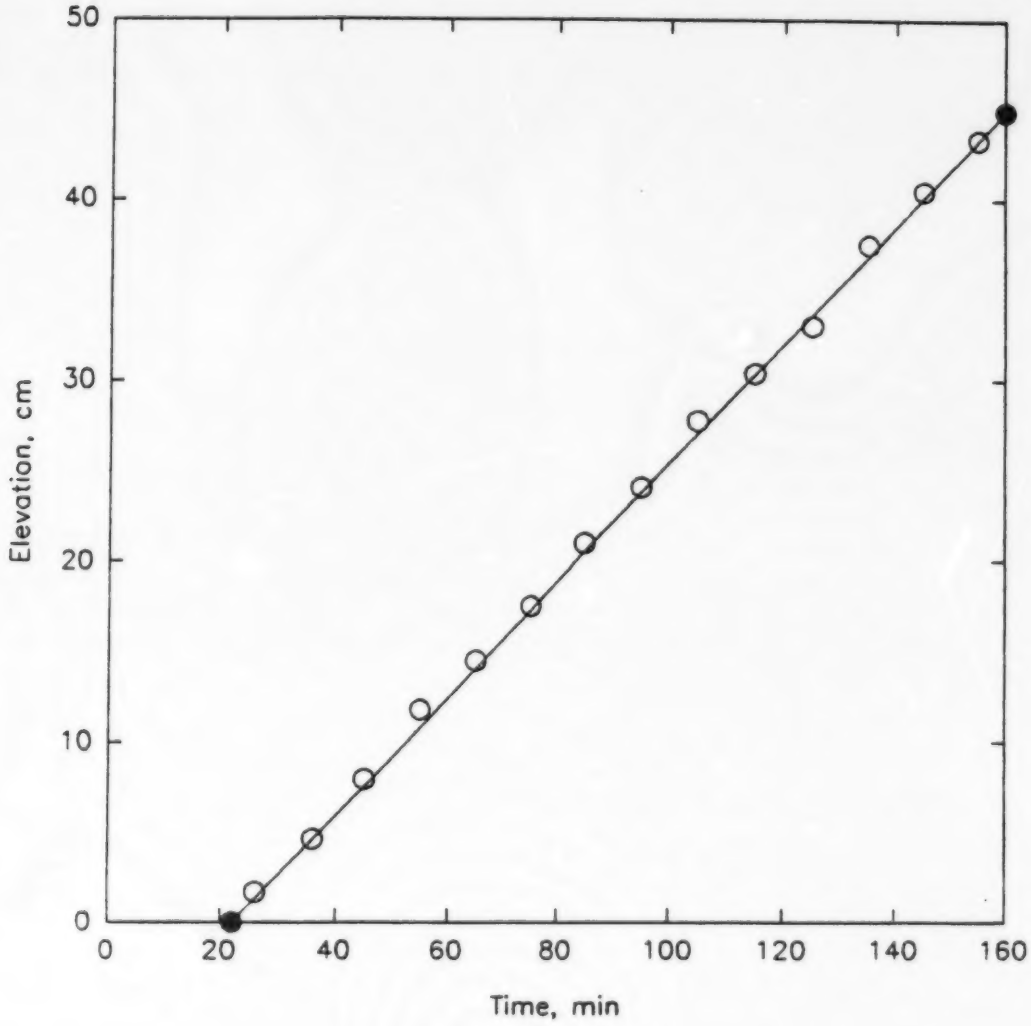

Figure 14 Rise of center of interface in Test #21. Bottom inlet employed with second screen mounted 3 cm above inlet. T_i = 20.8° C, T_0 = 1.5° C, V = 1.12 cm/s, a = 7.12 cm^2. Solid line is fitted to measurements, dashed line is Eq.(8).

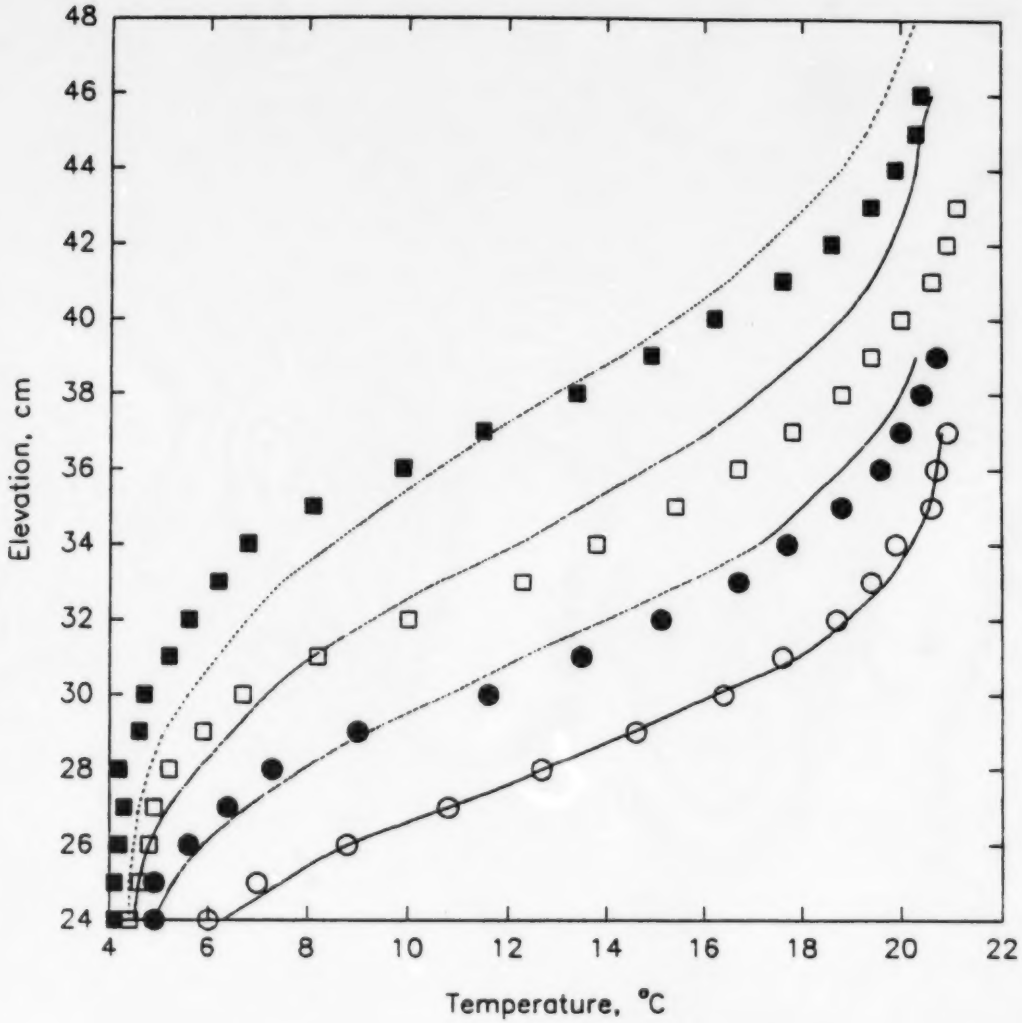

Figure 15 Temperature profiles in Test# 21 measured at four times. Open circles are t = 93 min, filled circles are t = 113 min, open squares are t = 123 min, and filled squares are t = 133 min. Solid line is error function fitted to the earliest profile at the center of the interface. Dotted lines are error function profiles which would result from the combination of upflow and conduction. Profiles are actually thinner than would be the case if conduction were the only heat transport.

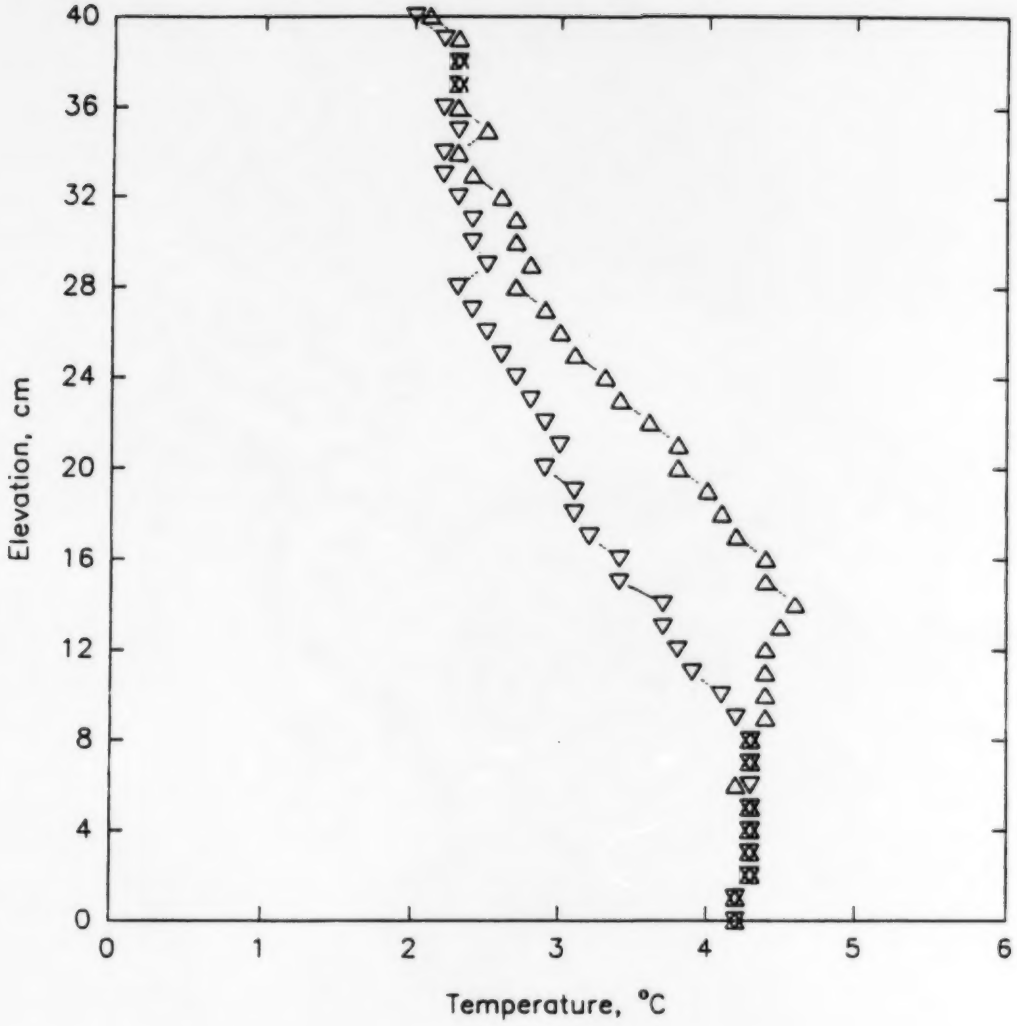

Figure 16 Temperature profiles measured in test #24 for which inflow was from top at 1.2° C into T_i = 4.4° C. Inflow at Q = 922 cm through grid. Triangles up and triangles down represent profiles taken at t = 73 min and 82 min respectively. Elevation of water particles moving at Q/A is 38 cm at 73 min. Mixing accounts for descent of interface below this elevation.

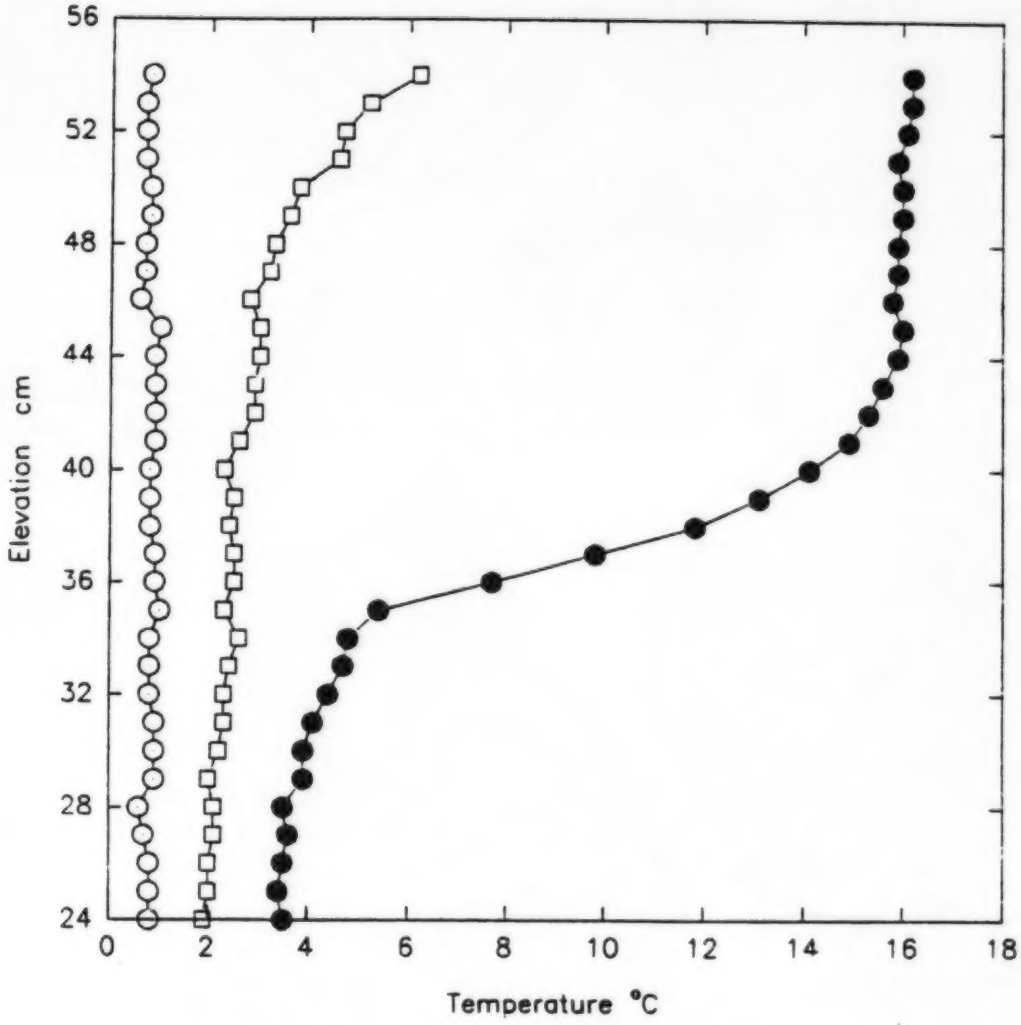

Figure 17 Temperature profiles measured in test #26 which simulated the filling with warm water, T_0 = 15.3° C, of a tank of water at T_i = 0.8° C. Open circles are initial profile, open squares t = 25 min; filled circles t = 65 min.